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Rec'd June 25/73

## ROBERT NAPIER.

(1791—1876.)

*President* 1863—1865.

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Robert Napier was born at Dumbarton on the 18th June 1791. Having served an apprenticeship of five years as blacksmith with his father, he worked as a blacksmith and mechanic in Edinburgh and Glasgow. In 1815 he started on his own account in Glasgow as a blacksmith, and later he became an ironfounder and engineer, and built his first marine engine for the "Leven" steamboat in 1823. Five years later he removed to larger premises, subsequently adding a shipbuilding yard at Govan. Many first-class steamers of all sizes were built there, both by himself and the subsequent firm of Robert Napier and Sons. He was early connected with steam navigation, being associated in 1830 with the City of Glasgow Steam Packet Company, most of whose vessels were engined by him. In 1839 he helped to establish the Cunard Line of mail steamers plying between this country and North America. He also built in 1856 H.M.S. "Erebus," the first of the armour-clad vessels ordered for the British Navy. He died on the 23rd June 1876.

Mr. Napier became a Member of this Institution in 1856, and was President in 1863, 1864, and 1865.

R. Napier

# THE INSTITUTION

OF

# MECHANICAL ENGINEERS.

ESTABLISHED 1847.

## PROCEEDINGS.

1912.

PARTS 3–4.



PUBLISHED BY THE INSTITUTION,

STOREY'S GATE, ST. JAMES'S PARK, LONDON, S.W.



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1912.

PARTS 3–4.

# The Institution of Mechanical Engineers.

## PAST-PRESIDENTS.

George Stephenson, 1847–48. (*Deceased* 1848.)

Robert Stephenson, F.R.S., 1849–53. (*Deceased* 1859.)

Sir William Fairbairn, Bart., LL.D., F.R.S., 1854–55. (*Deceased* 1874.)

Sir Joseph Whitworth, Bart., D.C.L., LL.D., F.R.S., 1856–57, 1866. (*Deceased* 1887.)

John Penn, F.R.S., 1858–59, 1867–68. (*Deceased* 1878.)

James Kennedy, 1860. (*Deceased* 1886.)

The Right Hon. Lord Armstrong, C.B., D.C.L., LL.D., F.R.S., 1861–62, 1869. (*Deceased* 1900.)

Robert Napier, 1863–65. (*Deceased* 1876.)

John Ramsbottom, 1870–71. (*Deceased* 1897.)

Sir William Siemens, D.C.L., LL.D., F.R.S., 1872–73. (*Deceased* 1883.)

Sir Frederick J. Bramwell, Bart., D.C.L., LL.D., F.R.S., 1874–75. (*Deceased* 1903.)

Thomas Hawksley, F.R.S., 1876–77. (*Deceased* 1893.)

John Robinson, 1878–79. (*Deceased* 1902.)

Edward A. Cowper, 1880–81. (*Deceased* 1893.)

Percy G. B. Westmacott, 1882–83.

Sir Lowthian Bell, Bart., LL.D., F.R.S., 1884. (*Deceased* 1904.)

Jeremiah Head, 1885–86. (*Deceased* 1899.)

Sir Edward H. Carbutt, Bart., 1887–88. (*Deceased* 1905.)

Charles Cochrane, 1889. (*Deceased* 1898.)

Joseph Tomlinson, 1890–91. (*Deceased* 1894.)

Sir William Anderson, K.C.B., D.C.L., F.R.S., 1892–93. (*Deceased* 1898.)

Sir Alexander B. W. Kennedy, LL.D., F.R.S., 1894–95.

E. Windsor Richards, 1896–97.

Samuel Waite Johnson, 1898. (*Deceased* 1912.)

Sir William H. White, K.C.B., LL.D., D.Sc., F.R.S., 1899–1900.

William H. Maw, LL.D., 1901–02.

J. Hartley Wicksteed, 1903–04.

Edward P. Martin, 1905–06. (*Deceased* 1910.)

T. Hurry Riches, 1907–08. (*Deceased* 1911.)

John A. F. Aspinall, 1909–10.

# The Institution of Mechanical Engineers.

## OFFICERS.

1912.

### PRESIDENT.

| | |
|---|---|
| Edward B. Ellington, | London. |

### PAST-PRESIDENTS.

| | |
|---|---|
| John A. F. Aspinall, | Manchester. |
| Sir Alexander B. W. Kennedy, LL.D., F.R.S., | London. |
| William H. Maw, LL.D., | London. |
| E. Windsor Richards, | Caerleon. |
| Percy G. B. Westmacott, | Ascot. |
| Sir William H. White, K.C.B., LL.D., D.Sc., F.R.S., | London. |
| J. Hartley Wicksteed, | Leeds. |

### VICE-PRESIDENTS.

| | |
|---|---|
| Henry Davey, | Ewell. |
| Sir H. Frederick Donaldson, K.C.B., | Woolwich. |
| Michael Longridge, | Manchester. |
| The Right Hon. Lord Merthyr, G.C.V.O., | Aberdare. |
| A. T. Tannett-Walker, | Leeds. |
| Professor W. Cawthorne Unwin, LL.D., F.R.S., | London. |

### MEMBERS OF COUNCIL.

| | |
|---|---|
| William H. Allen, | Bedford. |
| Professor Archibald Barr, D.Sc., | Glasgow. |
| Sir J. Wolfe Barry, K.C.B., LL.D., F.R.S., | London. |
| George J. Churchward, | Swindon. |
| Dugald Clerk, D.Sc., F.R.S., | London. |
| Sir Robert A. Hadfield, D.Sc., D.Met., F.R.S., | Sheffield. |
| H. S. Hele-Shaw, LL.D., D.Sc., F.R.S., | London. |
| Edward Hopkinson, D.Sc., | Manchester. |
| J. Rossiter Hoyle, | Sheffield. |
| George Hughes, | Horwich. |
| Henry A. Ivatt, | Hayward's Heath. |
| Robert Matthews, | Manchester. |
| Donald B. Morison, | Hartlepool. |
| Sir Gerard A. Muntz, Bart., | Birmingham. |
| Eng. Vice-Admiral Sir Henry J. Oram, K.C.B., F.R.S., | London. |
| William H. Patchell, | London. |
| The Right Hon. Lord Pirrie, K.P., P.C., LL.D., | Belfast. |
| Walter Pitt, | Bath. |
| Mark H. Robinson, | London. |
| Captain H. Riall Sankey, R.E., ret., | London. |
| Wilson Worsdell, | Ascot. |

### HON. TREASURER.

Arthur Huson, F.Inst.B.

### AUDITOR.

Robert A. McLean, F.C.A.

### SECRETARY.

Edgar Worthington,
*The Institution of Mechanical Engineers,*
*Storey's Gate, St. James's Park, London, S.W.*
Telegraphic Address:—*Mech, Parl, London.* Telephone:—*Victoria*, 4564.

# The Institution of Mechanical Engineers.

## PROCEEDINGS.

### July 1912.

The Summer Meeting of the Institution was held in Belfast, commencing on Tuesday, 30th July 1912, at Ten o'clock a.m. The President, Edward B. Ellington, Esq., took the Chair in the Hall of the Municipal Technical Institute, after the Council and Members of the Institution had been welcomed by the Lord Mayor of Belfast, Councillor R. J. M'Mordie, M.A., M.P., and the Members of the Belfast Reception Committee.

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The Lord Mayor, in welcoming the President, the Council, and the Members of the Institution, said he presumed a large number of those present had already been welcomed individually in the City, and it was now his privilege to extend a formal but cordial welcome on behalf of the citizens as a whole. Belfast was favoured occasionally by visits of various organizations from the other side of the Channel, whether textile or engineering, while they had frequent visits of Learned Societies. The citizens were very glad to have such visits, and he trusted that the City always acquitted itself as it should do when it was favoured with the presence of distinguished strangers. On the present occasion the welcome was particularly hearty, because although all Belfast people were not engaged in engineering work they always seemed to be more or less in touch with it. He thought the community as a whole might be looked upon as highly mechanical. In the

(The Lord Mayor of Belfast.)

surrounding district there existed men who, although they had never received any technical training, could do many wonderful things in the mechanical line. He knew places in the country where farmers and their sons could build a house, doing all the masonry work and the joinery work in their spare time. The community as a whole was mechanical, and possibly they ought to have more mechanical works in the City than, as a matter of fact, existed. The Members would understand, however, that Belfast had to import all its raw material, and the engineering firms therefore could not compete on equal terms in the markets of the world when dealing with very heavy castings or machinery. The Belfast manufacturers confined themselves largely, he believed, more or less to manufacturing things in which the proportion of skilled labour to the raw material was very large. He would not, however, discuss a subject with which he was not very familiar.

While expressing the great pleasure the community enjoyed in receiving such a distinguished body as the Institution of Mechanical Engineers, he pointed out that there was one drawback in connection with the Meeting, namely that a sufficient number of members had not come to Belfast. The citizens had been expecting nearly double the number that had actually come, and he hoped that, when those who had attended left the City, they would be able to report to the members who had not come that they had made a mistake in staying at home. He knew that the community as a whole was delighted with the visit, and he was convinced that the feeling would receive due expression during the coming week. He was confident that their deliberations would be successful, that nothing but goodwill would exist during the Meeting, and that very pleasant memories would be left when they had gone.

The PRESIDENT, on behalf of himself as President of the Institution, of the Council, and of the Members, thanked the Lord Mayor most heartily for the kind words of welcome to which he had just given expression. It was quite true that in Belfast and its immediate neighbourhood there was an immense amount

of mechanical interest. When the Institution paid its summer visits, which were such an important part of the Institution's work, they went where they knew they could learn something which they could not learn at home. The Members wished to broaden their ideas, and to keep in touch with all that was best in the country. Mechanical engineers were cosmopolitan; in fact, all engineering was cosmopolitan, and there was also a bond of union wherever they went among the members of the Institution. Within the last eight years the Institution had paid Summer Visits in England on three occasions, also a Meeting in Scotland, in Wales, in Belgium, in Switzerland, and now in Ireland; one of the Meetings held in England was a joint meeting with the American Society of Mechanical Engineers. It would thus be seen that their sympathies were broad. He would not say any more at the moment, except to express again the very hearty thanks of the Members to his Lordship for his kind welcome.

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The President said he very much regretted to have to formally announce to the Members what most of them already knew, namely, that during the last week the Institution had lost by death their old friend, Henry Lea, of Birmingham. The late Mr. Lea was a Member of the Institution for a great many years, namely, since 1860. He served for a long time upon the Council, and was most highly esteemed by them. He was sure the Members would unanimously agree that a letter of condolence should be sent on behalf of the Meeting to his surviving relatives.

The Resolution was carried in silence, all the Members upstanding.

The Minutes of the previous Meeting were read and confirmed.

The President announced that the Ballot Lists for the election of New Members had been opened by a Committee appointed by the Council, and that the following two hundred and six candidates were found to be duly elected:—

MEMBERS.

| | |
|---|---|
| Bayfield, Henry Arthur, | Vancouver, B.C. |
| Caspersen, Hans Einar, | Glasgow. |
| Catto, John, | Newcastle-on-Tyne. |
| Colclough, Ernest, | Oswestry. |
| Coley, Arthur Boden, | London. |
| Cook, Thomas, | Stalybridge. |
| Couper, John Duncan Campbell, | Newport, Mon. |
| Dudgeon, Harry Allsop, | Glasgow. |
| Duncan, Edward MacGregor, | London. |
| Fairlie, William Edward, | London. |
| Goudie, Robert, | Johnstone. |
| Hall, William, | Preston. |
| Handoll, Henry, | London. |
| Hendry, Robert Bell, | Cape Town. |
| Hutton, William Mack, | Bulawayo. |
| Jarmain, Edward Albert, | Leeds. |
| Lacey, Edwin Charles, | London. |
| Lake, William Osborne, | London. |
| Lees, William, | Manchester. |
| Lewis-Heath, Frederick John, Eng. Lieut., R.N., | H.M.S. "Itchen." |
| McClean, William Newsam, | London. |
| Macdonald, William Richard, | London. |
| McGregor, Stuart William Buchanan, | London. |
| MacKirdy, John Hansard, | Calcutta. |
| Marshall, Frederick William, Eng. Commander, R.N., | London. |
| Mertens, John Thompson, | Calcutta. |
| Paramor, John, | Watford. |
| Paterson, Alexander, | London. |
| Pilling, James, | Manchester. |
| Plowright, John Tebbutt, | Hull. |
| Prout, Frank, | Manchester. |
| Reid, James, | Glasgow. |
| Sanguinetti, Vivian, | Tokyo. |

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|---|---|
| Tookey, William Alfred, . . . | London. |
| Wheatley, William George, . . . | Calcutta. |

ASSOCIATE MEMBERS.

| | |
|---|---|
| Abell, Charles Edmund, . . . | Worcester. |
| Adams, James Walter, . . . . | Bradford. |
| Allman, Arthur James Rochfort, . . | Isleworth. |
| Anderson, John, . . . . . | Glasgow. |
| Armstrong, Percy Towns, . . . | Leeds. |
| Ash, Wilfrid Cracroft, . . . . | Calcutta. |
| Bacon, James, . . . . . | London. |
| Bailey, David Edward, . . . . | Charlton. |
| Bailey, Henry Charles, . . . . | Hillcrest, Alberta. |
| Ball, William Francis, . . . . | Torquay. |
| Bamford, Daniel, . . . . . | Manchester. |
| Beech, Albert, . . . . . | Smethwick. |
| Bentall, Anthony Frank, . . . | London. |
| Blackwell, George, . . . . | Sheffield. |
| Blakeney, Stepney Edward, . . . | London. |
| Brotherton, Eric, . . . . . | London. |
| Burdock, Arthur, . . . . . | Southampton. |
| Cameron, Douglas Evan, . . . . | Cardiff. |
| Campbell, John Donald, . . . . | Howrah. |
| Cave-Browne-Cave, Thomas Reginald, Eng. Lieut., R.N., . . . . . | London. |
| Chalmers, Alexander Spence, . . . | Nyasaland. |
| Cockcroft, Ernest Egbert, . . . | Manchester. |
| Cragg, William Oliver, . . . . | Manningtree. |
| Craig, Wesley, . . . . . | Manchester. |
| Davy, Edward Vinicombe, . . . | San Juan, Arg. Rep. |
| Dickinson, Herbert, . . . . | Huddersfield. |
| Dickson, Ernest Holmes Llewelyn, . . | Salford. |
| Dixon, Ralph Claiborne, . . . | Canton. |
| Donald, Andrew, . . . . . | Calcutta. |
| Donkin, Herbert Julyan, . . . | London. |
| Donnelly, Samuel Holt, . . . | Letchworth. |

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|---|---|
| Dwyer, David Anthony, . . . . | London. |
| Edwards, Thomas Harold, . . . | London. |
| Everett, Edwin John Eustace, . . | Sclessin-Liége. |
| Fowler, Charles Henry, . . . . | Leeds. |
| Gamble, Stuart Arthur, . . . . | Manchester. |
| Gibbs, John Edward, . . . . | York. |
| Goddard, William Cory, . . . . | Hassocks, Sussex. |
| Graham, Ernest, . . . . . | Pernambuco. |
| Green, Ronald, . . . . . | London. |
| Grimwood, Bertie Constantine Ruffell, . | London. |
| Hanson, James, . . . . . | Penrhyndeudraeth. |
| Hardman, James Herbert, . . . | Manchester. |
| Harris, Vincent George, . . . | Manchester. |
| Harrison, George, . . . . . | Bilbao. |
| Hart, William Barnard, . . . | London. |
| Haydon, Gerald Summersell, . . . | Wabana, N.F. |
| Hind, Harold Robert, . . . . | London. |
| Hollingsworth, Frederick, . . . | Cape Town. |
| Hopkinson, Arthur, . . . . | Chesterfield. |
| Horsburgh, George Donald Lee, . . | Manchester. |
| Hughes, Frederick, . . . . | Cardiff. |
| Hutchinson, George Victor Valentine, . | Dublin. |
| Innes, William, . . . . . | London. |
| Jantzen, Paul Hermann Hudden, . . | London. |
| Johnson, Arnold Robert, . . . | Penang. |
| Kendall, Richard, . . . . | London. |
| Kingston, Charles Sydney, . . . | Kilmarnock |
| Lambourne, Albert, . . . . | Brighton. |
| Langton, Stephen Wickham Bartlett, . | London. |
| Lawrence, John James Rutter, . . | Mexico. |
| Lea, James Thomas, . . . . | Durban. |
| Leadbeater, William, . . . . | Gainsborough |
| Lofthouse, William, . . . . | Wakefield. |
| MacKinnon, Alister, . . . . | Buenos Aires. |
| Mann, Frederick Cyril Duncombe, . . | London. |
| Mawer, Sydney, . . . . . | Leiston, Suffolk. |

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|---|---|
| Megirian, Joseph James, | New York. |
| Mein, James Henry, | Grimsby. |
| Millar, Alexander, | Kilmarnock. |
| Mountney, Charles Frederick, | Wallsend-on-Tyne. |
| Mulligan, Michael, | Bolton. |
| Munro, Carlos Sinclair Adam, | Twickenham. |
| Murchison, Kenneth, | Havana. |
| Nash, Alfred William, | Mohammerah, Persian Gulf. |
| Neal, James Henry, | Teddington. |
| Nelson, Charles Cowley, | Hong Kong. |
| Newlands, Arthur Herbert, | Barrow-in-Furness. |
| Newman, Alexander Richard, | London. |
| Norman, Edmund Golledge, | Calcutta. |
| Oliver, Alfred, | Lincoln. |
| Penrose, Philip Owen, | Chatham. |
| Phillips, O'Moore Francis, | Ruabon. |
| Pollock, Charles Albert, | London. |
| Priestley, Charles Gordon, | Nottingham. |
| Quinton, Walter Richard, | Woolwich. |
| Reynolds, Charles Harold, | Portsmouth. |
| Reynolds, Edward William, | Portsmouth. |
| Richards, Francis Bartlett, | London. |
| Richards, Frank Herbert, | Woolwich. |
| Richardson, Harold Willoughby, | London. |
| Rossiter, Thomas Henry, | Bristol. |
| Saxelby, Fred Albert, | Loughborough. |
| Scoffham, Francis Bew, | Birmingham. |
| Scott, John MacDonald, | London. |
| Sheret, David Alexander, | Newcastle-on-Tyne. |
| Skidmore, Thomas Emmott, | Shanghai. |
| Smith, James Thomson, | Glasgow. |
| Sprague, Joseph Ernest, | Pontypridd. |
| Staniar, Henry Drummond, | Manchester. |
| Stevenson, William Lewis, | Calcutta. |
| Stirling, Andrew Gibson, | Demerara. |

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|---|---|
| Stratford, Aubrey Bertram, . . . | Manchester. |
| Swain, Herbert John, . . . . | Cambridge. |
| Tarrant, Arthur Norman, . . . | Poole. |
| Thompson, Harry James, . . . | Howrah. |
| Thompson, Jacob Jewett, . . . . | Canton. |
| Thompson, St. John Moriarty, . . . | London. |
| Ward, Frank Eldridge, . . . . | London. |
| Watkins, William Gordon, . . . | London. |
| Weiss, Maurice, . . . . . | London. |
| West, Frederick Buick, . . . . | Johannesburg. |
| Westmacott, Philip Guise, . . . | Bombay. |
| Wilkinson, Joseph William Frederick, . | Preston. |
| Wilkinson, Lionel St. George, . . | Rochdale. |
| Woodhouse, Ernest, Lieut. R.E., . . | Salisbury. |
| Young, John Robert, . . . . | Stafford. |

GRADUATES.

| | |
|---|---|
| Adams, Charles Henry, . . . . | London. |
| Barber, Alfred Thomas, . . . . | London. |
| Beevers, Leslie, . . . . . | Wakefield. |
| Bell, John, . . . . . . | Hamsterley, Co. Durham. |
| Bellman, Harold, . . . . . | London. |
| Bolton, Charles John Howard, . . | London. |
| Bond, Henry Fielding, . . . . | Leiston, Suffolk. |
| Brennand, John, . . . . . | Boston. |
| Cartwright, Ian Robert, . . . | Dumbarton. |
| Chrimes, Thomas Edward, . . . | Brighton. |
| Cook, John, . . . . . . | Birmingham. |
| Cook, John William Donald, . . . | Potters Bar. |
| Cromie, William Henry, . . . | London. |
| Cryer, Thomas Bond, . . . . | Bridgnorth. |
| Currie, Alan Whitmore, . . . | London. |
| Delves, Francis Joseph, Jun., . . . | London. |
| Druitt, Charles Lambert, . . . | London. |
| Duncan, Bernard Arthur, . . . | Liverpool. |

| | |
|---|---|
| Duthie, Harold, | West Hartlepool. |
| Eyles, George Frederic, | Erith. |
| Forward, Philip Grainger, | Coventry. |
| Gledhill, Charles, | Castleford, Yorks. |
| Ireland, Arthur James Thomas, | London. |
| Jackson, Charles Laurence Hamilton, | London. |
| Junner, Gordon Mackenzie, | London. |
| Lancaster, John Stuart, | Chesterfield. |
| Lewis, Arthur Ernest, | Falmouth. |
| Maize, William James, | Stoke-on-Trent. |
| Marshall, Fred, | Sleaford. |
| Mellanby, Thomas Graham, | Stockton-on-Tees. |
| Morris, John Turner, | London. |
| Nell, Walter Archibald, | London. |
| Norrish, Guy Ralph, | London. |
| Paddock, William George, | Birmingham. |
| Partington, James Sutcliffe, | Durban. |
| Power, Walter, | Liscard. |
| Prentis, Eric Francis, | Maidstone. |
| Reynolds, William Arthur, | London. |
| Riley, Thomas Sharp, | Huddersfield. |
| Roberts, Stanley Hugh, | London. |
| Robinson, Matthew Dolton, | Manchester. |
| Rolt, Frederick Henry, | London. |
| Sahgal, Sant Ram, | Edinburgh. |
| Saunders, Thomas Thornton, | London. |
| Skinner, Harold Vernon, | Lincoln. |
| Smith, Bernard Westcott Turner, | Swindon. |
| Stapylton-Smith, John Bryan, | Bexhill-on-Sea. |
| Stent, Arthur John, | Letchworth. |
| Stocks, John Arthur, | Huddersfield. |
| Swinton, Ernest, | Widnes. |
| Tindle, Stuart Pike, | Junin, Arg. Rep. |
| Wells, George Maurice, | Winchester. |
| Whitaker, Lionel Bertie, | St. Leonards-on-Sea. |
| Woodcock, Francis Stanley, | Bradford. |

The PRESIDENT announced that the following twenty-three Transferences had been made by the Council:—

*Associate Members to Members.*

| | |
|---|---|
| BLACKBURN, ALBERT ARTHUR, . . . | Belfast. |
| BORNER, OTTO LEO, . . . . . | London. |
| BUCHANAN, WILLIAM ERNEST, . . . | Simla. |
| BURY, RONALD EDWARD, . . . . | Lillooah. |
| CASSON, WILLIAM, . . . . . | London. |
| CRYER, JAMES WILFRED, . . . . | Bolton. |
| DENNISS, ARTHUR WORKMAN, . . . | Dundalk. |
| HOWARD, HENLEY LIONEL, . . . . | Barking. |
| LANGTON, JOHN MONTAGUE ELLIS, . . | Tantah. |
| LE MASURIER, JAMES, . . . . | Singapore. |
| LYONS, ROBERT ERNEST BRABAZON, . . | Entre Rios. |
| MARTY, ERNESTO, . . . . . | Peñarol. |
| METCALF, ALFRED TOWNLEY, . . . | Bradford. |
| MITCHELL, JAMES, . . . . . | Whitby. |
| PARKINSON, BERNARD ROBERT, . . . | London. |
| PURNELL, WALTER HENRY, . . . . | Loughborough. |
| REMINGTON, ALFRED ARNOLD, . . . | Birmingham. |
| SMITH, LOUIS WILLIAM, . . . . | Lincoln. |
| STONE, SIDNEY CHARLES EVE, . . . | London. |
| WALFORD, FREDERICK, . . . . | Bankipur. |
| WILLANS, GEORGE HERBERT, . . . | Smyrna. |
| WILSON, WILLIAM BUXTON, . . . | Penang. |

*Graduate to Member.*

| | |
|---|---|
| YERBURY, FREDERICK AUGUSTUS, . . . | Vancouver, B.C. |

The following Papers were read in abstract and discussed:—

"Rolling-Stock on the principal Irish Narrow-Gauge Railways"; by R. M. Livesey, *Member*, Locomotive Superintendent, Co. Donegal Railways Joint Committee, Stranorlar.

"New Graving Dock, Belfast: Mechanical Plant and General Appliances"; by W. Redfern Kelly, Engineer-in-Chief to the Belfast Harbour Commissioners.

"The Evolution of the Flax Spinning Spindle"; by John Horner, of Belfast.

At Half-past Twelve o'clock p.m. the Meeting was adjourned to the following morning.

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The Adjourned Meeting was held in the Hall of the Municipal Technical Institute, Belfast, on Wednesday, 31st July 1912, at Ten o'clock a.m.; Edward B. Ellington, Esq., President, in the Chair.

The following Papers were read in abstract and discussed:—

"Wire Ropes for lifting appliances, and some Conditions that affect their Durability"; by Daniel Adamson, *Member*, of Hyde.

"Reciprocating Straight-Blade Sawing-Machines"; by Charles Wicksteed, *Member*, of Kettering.

"Commercial Utilization of Peat for Power Purposes"; by H. V. Pegg, of Belfast.

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The PRESIDENT said the very pleasant duty devolved upon him of asking the members to pass a vote of thanks to the various gentlemen and firms who had contributed to the success of the Meeting at Belfast. He, therefore, had much pleasure in moving:—

"That the best thanks of the Members of the Institution of Mechanical Engineers, in this Meeting assembled, be given:—

To the Right Hon. the Lord Mayor of Belfast, Councillor R. J. M'Mordie, M.A., M.P., for his Welcome of the President, Council, and Members of the Institution to the City of Belfast; also to his Lordship and the Lady Mayoress, for their kind invitation to a Reception in the City Hall.

To the Belfast Corporation Library and Technical Instruction Committee, for the loan of the Hall and other rooms of the Municipal Technical Institute; and to Mr. Francis C. Forth, the Principal, for arranging other facilities connected with the Meetings.

To the Chairman of the Reception Committee, the Lord Mayor; the Vice-Chairmen, The Most Hon. the Marquess of Londonderry, K.G., P.C., G.C.V.O., The Right Hon. the Earl of Shaftesbury, K.P., K.C.V.O., Mr. J. Milne Barbour, D.L., Mr. R. H. Reade, D.L., and Mr. Robert Thompson, D.L., M.P.; the Honorary Treasurer, Mr. H. Incledon Johns; and Members of the Belfast Reception Committee, for the many arrangements they have made for the entertainment of the Members and Ladies.

To the Chairman and Directors of Messrs. Harland and Wolff, for their kindness in entertaining the Members to Luncheon.

To Mr. and Mrs. S. C. Davidson, for inviting the Members and Ladies to a Garden Party at Seacourt, Bangor.

To the Belfast Harbour Commissioners, for inviting the Members and Ladies to a Steamboat Excursion on Belfast Lough; also to Mr. Robert Thompson, M.P., Chairman

of the Harbour Commissioners, for inviting the party to Tea.

To the Belfast Corporation Improvement Committee, for the use of Ulster Hall for the Institution Dinner and the Institution Luncheon.

To Messrs. Harland and Wolff, the Belfast Harbour Commissioners, Messrs. Workman, Clark and Co., and the Proprietors of Places of Engineering Interest, for their kindness in throwing open their Works for the Visits of Members; also to the Royal Belfast Golf Club, the Royal County Down Golf Club and the Royal Portrush Golf Club, for the extension of hospitable facilities.

To the Chairman and Directors of the Midland Railway (Northern Counties Committee) and of the Belfast and County Down Railway, for their generous arrangements for travelling facilities.

To the Right Hon. the Earl of Roden, for kindly permitting a Visit to Tullymore Park.

To the Directors of Provincial Cinematograph Theatres, for their invitation to the Members and Ladies to a Cinematograph Exhibition of Subjects of Engineering Interest.

To the London and North Western, the Midland, and the other Railway Companies of the United Kingdom, for special travelling facilities connected with the Meeting.

To Mr. William A. Traill, M.A. Ing., for arranging to conduct the Members and Ladies over Dunluce Castle and the Giant's Causeway.

To the Joint Honorary Local Secretaries, Mr. S. C. Davidson and Mr. Bowman Malcolm, also to Mr. A. Basil Wilson and Mr. J. G. Harris, for planning visits to places of interest in Belfast and neighbourhood, and for the admirable arrangements which their forethought and energy have provided for each day of the Meeting."

The Resolutions were carried by acclamation.

The Meeting terminated at One o'clock p.m

The Meeting was attended by 229 Members and 37 Visitors 90 Ladies were also present.

# ROLLING-STOCK ON THE PRINCIPAL IRISH NARROW-GAUGE RAILWAYS.

By R. M. LIVESEY, *Member*, Locomotive Superintendent, Co. Donegal Railways Joint Committee, Stranorlar.

Before taking up the subject of this Paper, the author wishes to digress briefly, while he recounts a few of the reasons for, and the relative advantages and disadvantages of, narrow-gauge railways.

Practically, the only reason for the construction of a narrow-gauge line is cheapness, and no doubt in certain cases a considerable saving can be effected. But if, as in many instances in Ireland, such railway has to be fully equipped, almost on the same lines as a broad-gauge railway, in order to comply with the somewhat onerous requirements of the Board of Trade, then there is very little to be gained from the point of view of economy. The author has in mind one narrow-gauge railway which cost £11,500 per mile, exclusive of rolling-stock, although there was no really heavy work involved in its construction. For all practical purposes, the only saving is in land, a narrower width being required, and this is comparatively small. No railway should be built of narrow-gauge if the cost will exceed £5,000 per mile, and then only if the proposed line will be for ever isolated from those of standard gauge, and the traffic is always likely to be small. It would be decidedly better to build a

"light" railway of the standard gauge. As the mileage of narrow-gauge lines in Ireland is no less than 525, of which nearly all is 3-feet gauge, it seems regrettable, now that they have come to stay that the majority of them were not linked up to form one large system. The whole of the stock might thus have been built to a uniform standard. No two lines have similar stock, nor would they be readily interchangeable; even the height of buffer centres varies in them all.

Coming now to the subject of the Paper from the rolling-stock point of view, the disadvantages of a narrow-gauge line altogether outweigh the advantages, if any; and they may be summarized as follows:—

(*a*) Steep gradients; due to following closely the contour of the country, in an effort to save money in banks and cuttings.

(*b*) Sharp curves due to similar causes.

(*c*) Greater overhang required, in order to provide reasonable accommodation, with consequent greater liability to overturn.

(*d*) Greatly reduced speeds.

(*e*) Great inconvenience and loss, owing to break of gauge when coming into contact with a line of the standard gauge.

(*f*) A certain amount of cramping of parts, and accompanied by reduced accessibility.

With regard to the first two items, it seems that very little thought was given to future working, when many of the lines were projected. A comparatively small additional outlay, in the first instance, employed in reducing grades and making easier curves, would have been repaid many times over by the savings effected in working. Heavy gradients and sharp curves are quite as objectionable from the point of view of working, on a narrow-gauge as on a broad-gauge railway. These difficulties are practically insurmountable after the line has been made; but, in the case of items (*c*) to (*f*), they can be greatly minimized by careful design and a little forethought in working out details.

In the matter of the break of gauge, the difficulties of transhipment, in the case of goods and parcels traffic, can be greatly reduced by the employment of specially designed tranship-trucks, some of which have been in successful use on the County Donegal Railways for many years.

The Irish narrow-gauge lines appear to afford greater variety in the design of locomotives and other stock than do the broad-gauge railways; the general effect is pleasing and there is no doubt that, apart from all idiosyncracies of design and appearance, the stock referred to does excellent work under very trying conditions. The only limits to the size of narrow-gauge engines, etc., are those imposed by the weight of rail in use, permissible loads on existing bridges, and necessary clearance of existing structures. The gauge itself has little influence on the size of engine, if speed is restricted; the greater tendency to overturn on curves can be counteracted to a large extent by giving the outer rail ample super-elevation. This is relatively greater for the narrow-gauge than for the broad-gauge for the same speed. The comparatively high speed of 40 miles an hour is quite common on the more important narrow-gauge lines.

The type of engine almost universally adopted is the "side-tank." There are only two tender-engines in use, namely, on the Londonderry and Lough Swilly Railway; and the Ballycastle Railway have a couple of "saddle-tank" engines. In the design of the earlier narrow-gauge lines, parts were frequently cramped and inaccessible, but, with experience and confidence in their possibilities, these defects are disappearing. One of the difficulties in the design of narrow-gauge engines is the fire-box, especially in the larger types. Owing to the very restricted space between the wheels, and to the fact that the length of fire-box is fixed by the conditions of firing, namely, the difficulty of properly distributing the coal at a distance from the fire-hole, it is not easy to get a grate area sufficient to burn the requisite amount of fuel. If all the coupled wheels could be placed in front of the fire-box, this difficulty would be disposed of; but, in practice, such a course is seldom feasible.

The form of the fire-box in cross-section differs materially from that of a standard-gauge engine; the design, spacing, and position

of side and roof stays require careful study and great experience, if trouble is to be avoided. Generally speaking, enough attention has not been paid to this subject, one designer merely copying another, and consequently many such fire-boxes are a continual source of anxiety to those in charge. The copper plate, in contact with the fire, usually suffers more severely in a narrow-gauge engine than in a broad-gauge, owing to this restricted grate area and to the greater blast-pressure required in order to burn the necessary quantity of fuel in a limited time, with consequent severer scouring action on the plate surfaces. This involves higher temperatures, which have a serious effect upon the plates in a shorter time than would be the case on a standard-gauge engine. Tube-plates suffer more severely from the same cause, and also from the greater frequency of the changes of temperature, as well as the greater range, which produce severer and more rapid reversals of stresses.

The author has had great experience of fire-box troubles with narrow-gauge engines, and when abroad he experimented with Low Moor iron and steel fire-boxes, with the result that the latter gave a considerably longer life than iron, and the iron than copper, where the water was exceptionally bad. All fire-box troubles were ultimately eliminated by the introduction of a circular steel fire-box, which gave every satisfaction and was very much cheaper in first cost. With regard to fire-box design an interesting series of articles appeared in *The Engineer* * dealing with causes of failure of tube-plates, etc., which will repay careful study.

In Ireland, outside cylinders only are the rule, as there is no room between the frames, and usually the latter are outside the wheels. This gives greatly enhanced steadiness in running. A leading four-wheel bogie is very generally used, though where the load per axle does not exceed the maximum permitted by the weight of rail, there is no reason why a two-wheel bogie or pony truck should not be used—provision of course being made for lateral movement—and so reduce the non-effective weight of engine.

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* *The Engineer*, 23 February 1912, *et seq.*

The couplings in almost universal use are of the "central combined buffer and draw-hook" type; they are automatic couplers, very efficient and perfectly safe. Frequently a considerable amount of side-play is provided in order to allow freedom and flexibility on curves, and on carriages a "slack gathering" apparatus is usually fitted; though there is no such fitting which can be regarded as sufficiently reliable and effective in its action. The screw-coupling in use by the Cork, Blackrock and Passage Railway is certainly the best for this purpose. Experience, however, seems to show that a slack gathering apparatus is not absolutely essential, at any rate on narrow-gauge lines with the central buffer. Side coupling-chains, as an additional precaution, are generally used, but on the County Donegal Railways they are not coupled for service, except when the draw-hook will not engage, as is sometimes the case on sharp curves in station yards; and the side chains then enable the vehicles to be drawn on to the straight to allow of the buffer engaging properly.

There is little else in the design of narrow-gauge stock that presents any very special difficulties or that differs materially from the standard-gauge styles. In the working of the stock, particularly engines of the heavier type, more attention is required to be given to lubrication, and a larger quantity must be used in the case of narrow-gauge engines than in broad-gauge. It is generally and roughly assumed that a narrow-gauge engine, travelling at, say, 30 miles an hour, has its moving parts doing as much work as a standard-gauge locomotive, running at 60 miles an hour. An example from actual practice will make the point clearer. The figures on page 604 are taken from a typical engine of the broad- and narrow-gauges respectively.

Obviously it will require a greater quantity of oil to lubricate 540 square feet of surface than it would for only 407 square feet; similar reasoning applies to other working parts. Attention is drawn to this point, because it is often urged that narrow-gauge engines require considerably less oil than the standard engines. The difficulties of lubrication are enhanced, in the former case, by the greater proximity of the working parts to the ground and to

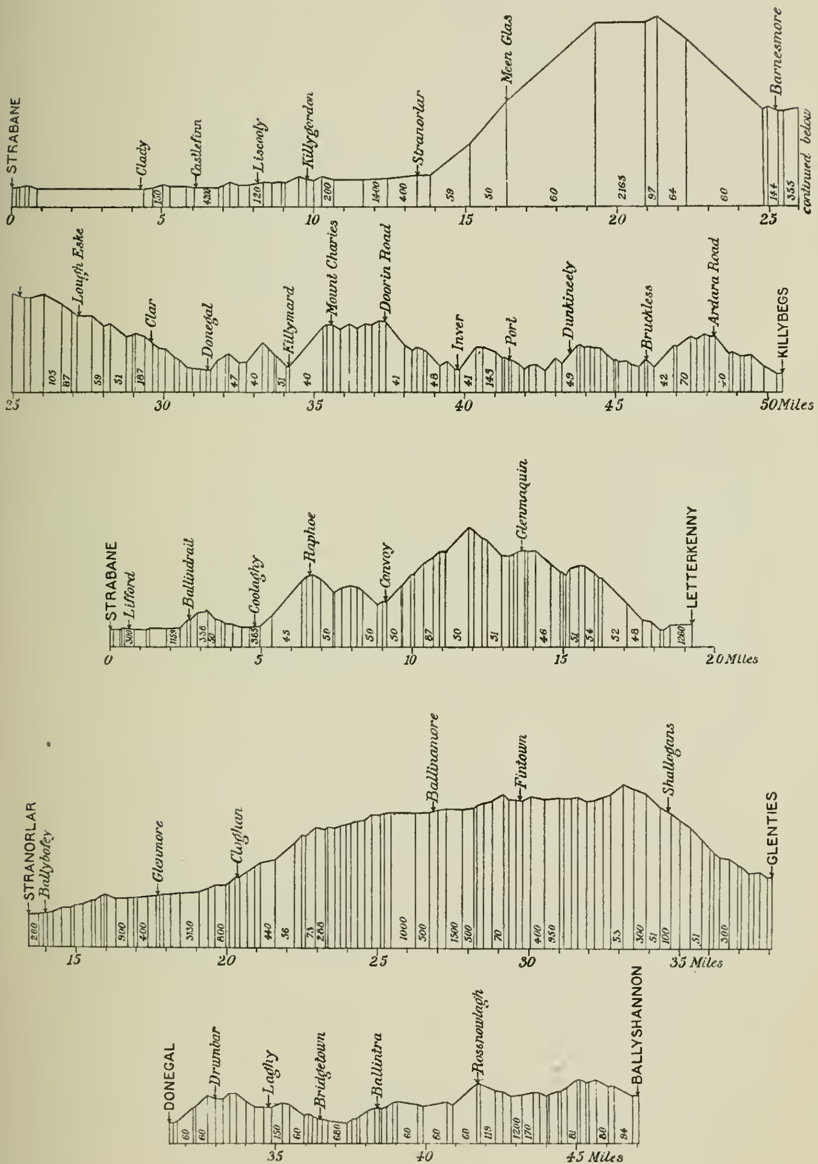
| —— | Broad-Gauge Engine. | Narrow-Gauge Engine. |
|---|---|---|
| Diameter of driving-wheels . . . . | 6 ft. 6 in. | 3 ft. 9 in. |
| Revolutions per mile . . . . . | $258\frac{1}{2}$ | 448 |
| Diameter of journals . . . . . | $8\frac{1}{2}$ in. | $8\frac{1}{2}$ in. |
| Length of journals . . . . . . | $8\frac{1}{2}$ in. | $6\frac{1}{2}$ in. |
| Pressure on journal (lb. per sq. in.) . . | 201 | 202 |
| Lineal movement of circumference of journal per mile . . . . . | 575 ft. | 997 ft. |
| Surface swept by journal per mile . . | 407 sq. ft. | 540 sq. ft. |

the consequent easier access of dust and grit to the working parts; this is especially the case in dry summer weather, and in some instances it is necessary to close in the motion work entirely, to protect it from injury.

It need hardly be stated that the power required to haul a given load is independent of the gauge, and the cost would be the same for any gauge. Any advantage in cost — to the narrower gauge, where it exists—is due to the relatively lesser tare weight of vehicles and slower speeds, and not to the gauge. On the larger and more important of the narrow-gauge lines in Ireland, the carriages and wagons will accommodate quite as much traffic of the same volume and weight as the broad-gauge vehicles, and in some cases more so, yet the tare of the former is considerably less. The wagons of the County Donegal Railways will all carry a net load of 7 tons, and the average tare of each is $3\frac{3}{4}$ tons, though all are equipped with both hand and vacuum automatic brakes; they are very substantially built and are capable of being run at any speed. The present standard six-compartment carriage of the same railway accommodates sixty people comfortably, is fully equipped with vacuum automatic brake, with acetylene lighting plant, etc., and weighs only 11 tons 14 cwt. or 3·9 cwt. per passenger.

FIG. 1.—*County Donegal Railways Joint Committee Lines.*

Profiles of Main Line and Three Branches.



For the reasons just enumerated, the narrow-gauge stock is relatively much more efficient than the broad-gauge; in other words, the proportion of paying to non-paying loads is considerably higher, and some authorities put this figure at from 12 to 15 per cent. The foregoing remarks are not put forward as showing the advantages of a narrow-gauge line, but serve to show where economy could be effected by judicious reductions in the scantlings and weights of broad-gauge vehicles.

The author proposes briefly to describe and illustrate by outline drawings and photographs the locomotives, carriages, and wagons in use on the more important narrow-gauge lines in Ireland.

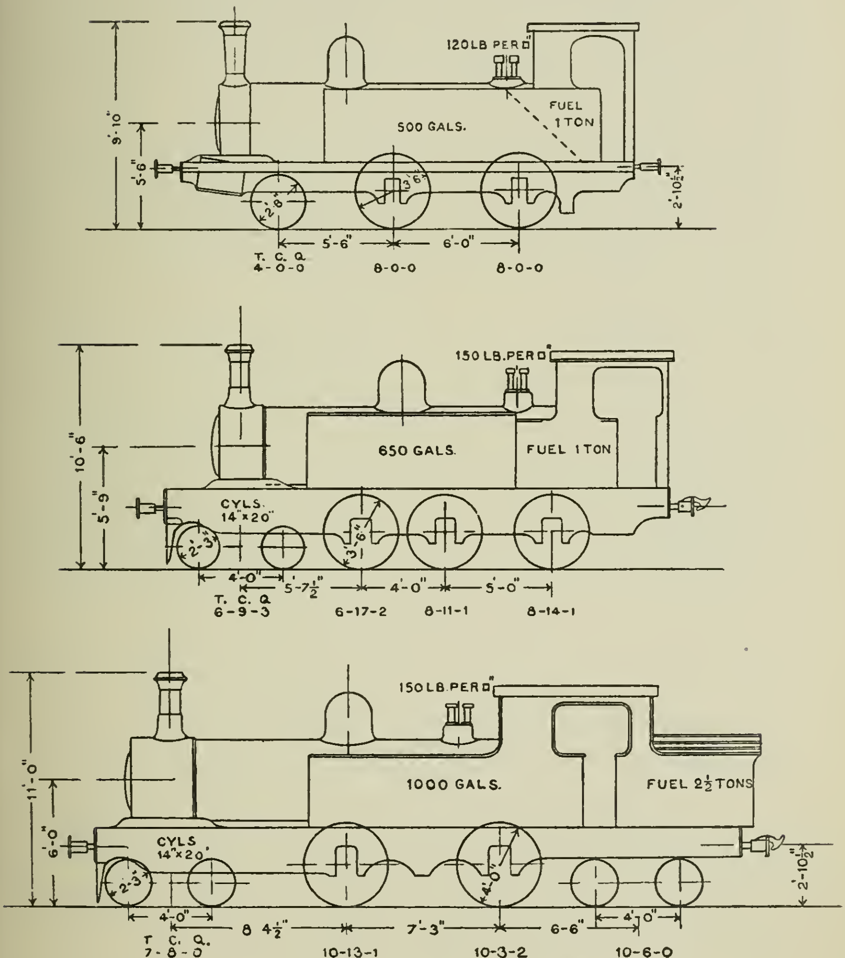
*County Donegal Railways Joint Committee.*—This is the longest and most important of the Irish narrow-gauge lines, the line itself being one of the most difficult in the country. It will be observed from the gradient diagram, Fig. 1 (page 605), that it is of a switchback character, and the curves are even more numerous than the gradients, the worst cases consisting of a $7\frac{1}{2}$-chain curve on a grade of 1 in 40; curves of 8 chains are common, but the average is about 10 chains. The stock of this Committee work over $125\frac{1}{2}$ miles of line, and it need hardly be said that some heavy hauling has to be performed. The traffic consists mainly of heavy goods, but all trains are mixed and have to keep good time, the average speed between stations exceeding 30 miles an hour. In Table 1 (pages 624–7), of comparative dimensions of engines, the loads given for this line are such as the engines will take on the grade mentioned at 15 miles per hour. This line possesses 21 engines, 56 carriages, and 315 wagons, and other special vehicles for traffic purposes; and the number of miles run per engine per annum, as compared with those of other Irish lines (narrow-gauge), is evidence of the amount of work done. The engines are divided into six classes, in the order in which they were introduced. Figs. 2–7 (pages 607–8) and Plate 23 illustrate them all.

Fig. 2 shows the original type of engine in use thirty-one years ago, one of which is still doing good work on branch lines and

Figs. 2, 3 and 4.

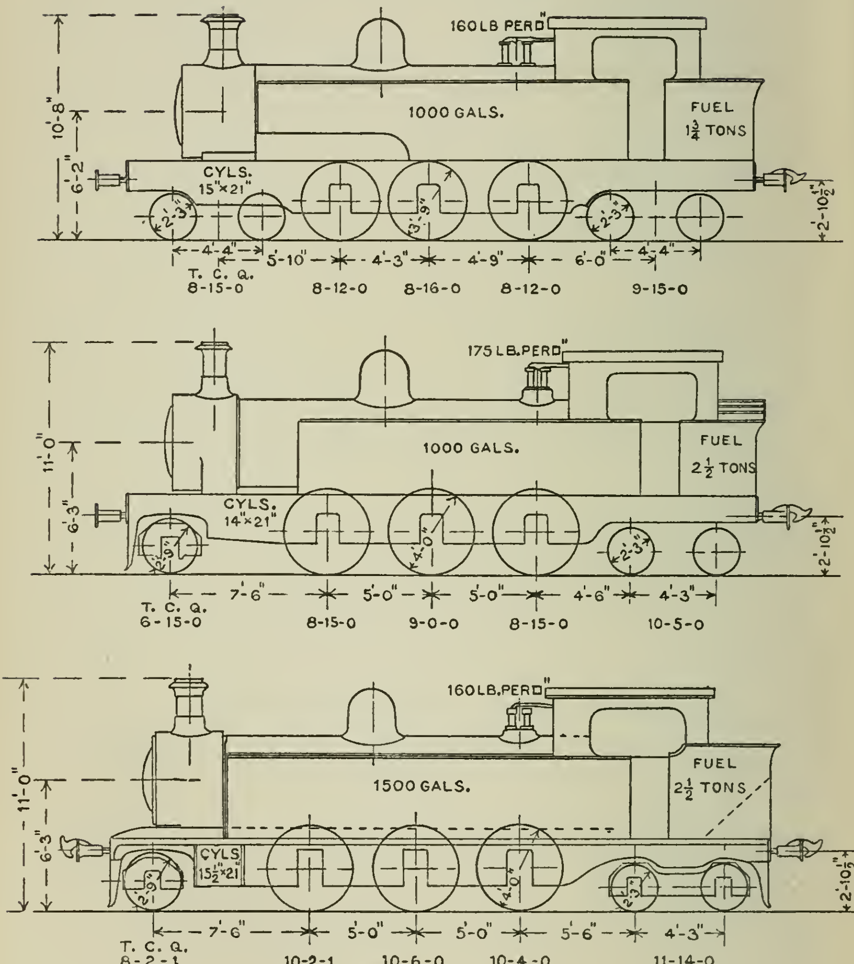
*County Donegal Railways.*

Cylinders 13 inches × 20 inches, and 14 inches × 20 inches.



FIGS. 5, 6 AND 7.

*County Donegal Railways.*



is also useful as a shunting engine. This was followed (on the opening of the "Balfour" lines in 1893) by the type shown in Fig. 3; these engines have done good service, and will deal with very heavy loads and are capable of very rapid acceleration. The water and coal capacities soon proved too small, frequent delays being due to this cause, and in Fig. 4, which followed, ample provision was made to get rid of this trouble; at the same time a return was made to the four-coupled type and a larger wheel substituted. This latter change was made as the engines were intended for fast passenger work between Stranorlar and Derry, after the line was extended from Strabane to the last-mentioned city in 1900. The loads on coupled wheels were then high for a narrow-gauge line, and in consequence the use of these engines was restricted to the section named, as on the Stranorlar to Donegal section the rails were only 45 lb. per yard, as compared with 60 and 65 lb. on the other section.

As the goods traffic grew a more powerful type of engine was found necessary, and 4–6–4, Fig. 5, was introduced; this engine has a Belpaire fire-box, largely increased heating surface, larger cylinders, and a smaller coupled wheels. These engines are capable of taking heavy loads, but have never been good steamers, and were extravagant in coal. They were also restricted to the section above named, but after replacing the 45 lb. rails with 60 lb. B.S.S. rails in 1907, all restrictions were removed.

The type shown in Fig. 4 having proved itself the most useful and economical engine, this was taken as the model for the design shown in Fig. 6. The same boiler was used, the cylinders were lengthened 1 inch, the boiler-pressure increased to 175 lb. per square inch, an extra pair of coupled wheels added and a radial truck substituted for the bogie in front. The result came up to all expectations, as this engine proved itself eminently capable of doing the same work as Fig. 5, at rather better speeds, and at greatly reduced expense. They do the same work on an average of 6 lb. of coal per mile less than Fig. 5; for the first year it was 10 lb. per mile less.

The necessity for dispensing with the watering of engines on the longer runs required a greater tank capacity, and this was

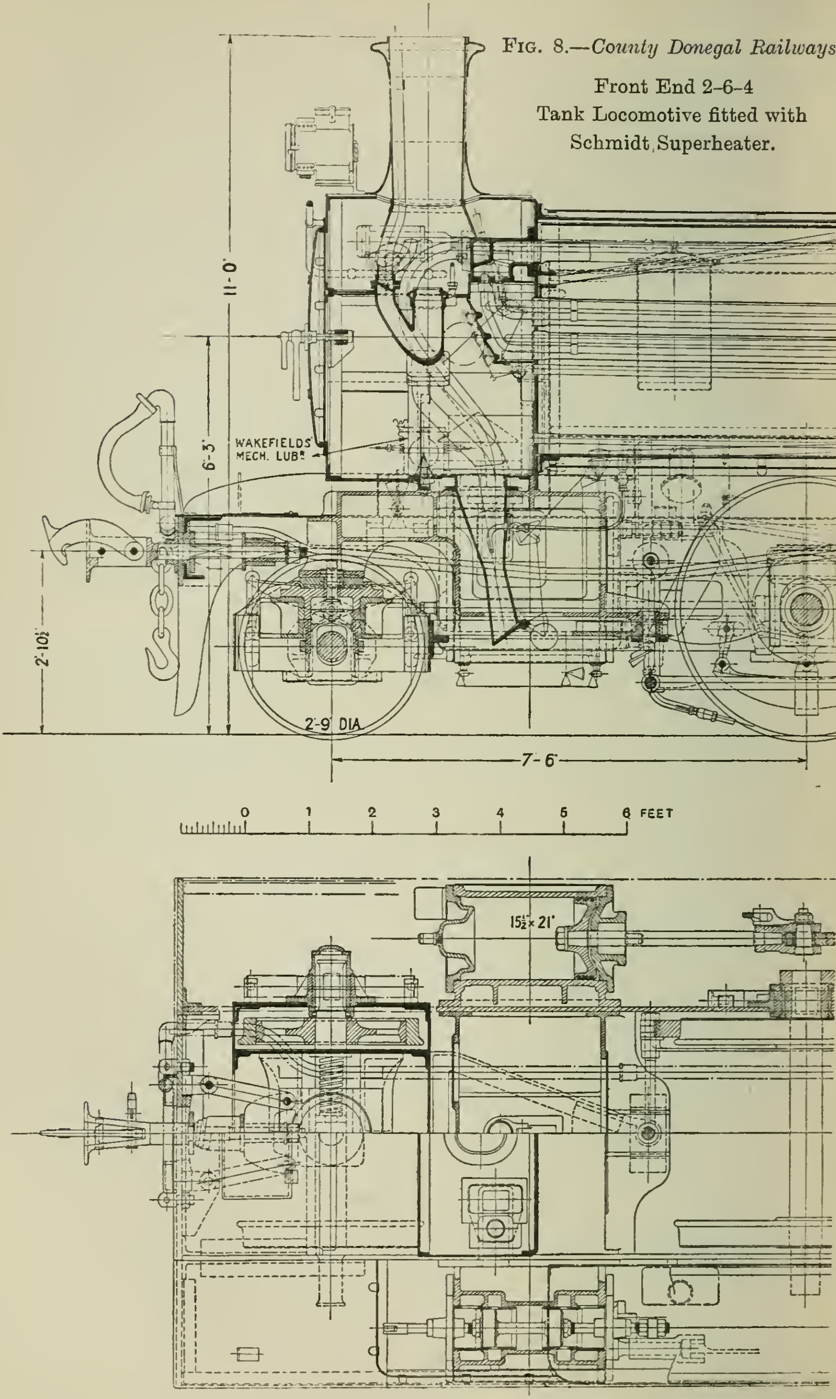
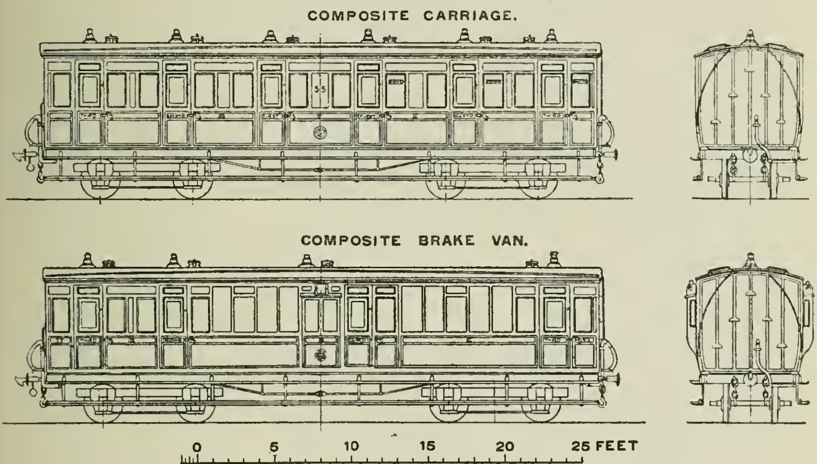



FIG. 8.—*County Donegal Railways.*

Front End 2-6-4 Tank Locomotive fitted with Schmidt Superheater.

provided for in the class shown in Fig. 7. It was originally intended to provide 2,000-gallon tanks, but on the suggestion of the builders, that by using Schmidt's superheater the capacity could be reduced to 1,500 gallons without interfering with the working radius of the engine, this proposal was adopted. The conditions of service on these lines are scarcely ideal for a superheater engine of this type, owing to the frequent stops and the necessity of shutting off steam so often on down grades. The engines have only just been put into work, so that it is not yet possible to judge of their

Figs. 9 and 10.—*County Donegal Railways.*



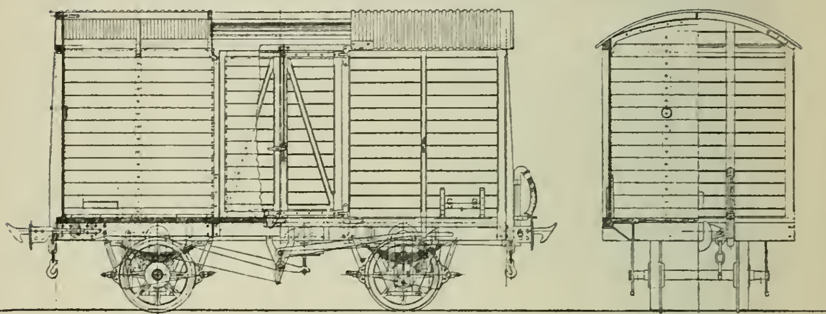

performance, but by the time this Paper is read the author may be in a position to give some particulars of their work. These engines are interesting in that they are the first narrow-gauge engines in the British Isles to be fitted with a superheater. They were originally intended to be duplicates of Fig. 6, excepting in tank capacity, but the adoption of the superheater, the larger cylinders, and the use of piston-valves, modified the type considerably, and they were classed as Fig. 7. The cylinder and valve lubrication is effected by Wakefield's mechanical lubricator; an automatic damper control cylinder is fitted, also Boyer speed-indicator and recorder; forced lubrication for all axle-journals and

bogie-centres, smoke-box ash-ejector, variable blast-pipe, also sundry minor improvements introduced to save labour and oils. Fig. 8 (page 610) shows the general arrangement of the front end of this engine.

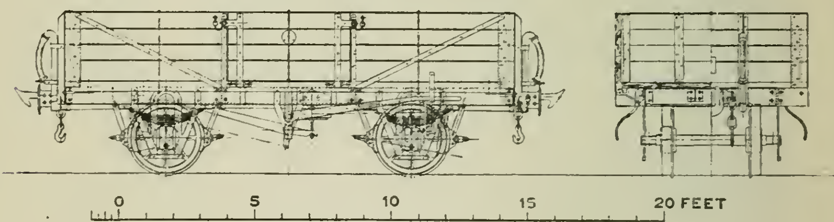
The carriages of the Donegal lines are very similar to those of the 4-foot 8½-inch gauge, and are quite as commodious as those

FIGS. 11 AND 12.—*County Donegal Railways.*

COMBINED COVERED GOODS AND CATTLE WAGON.



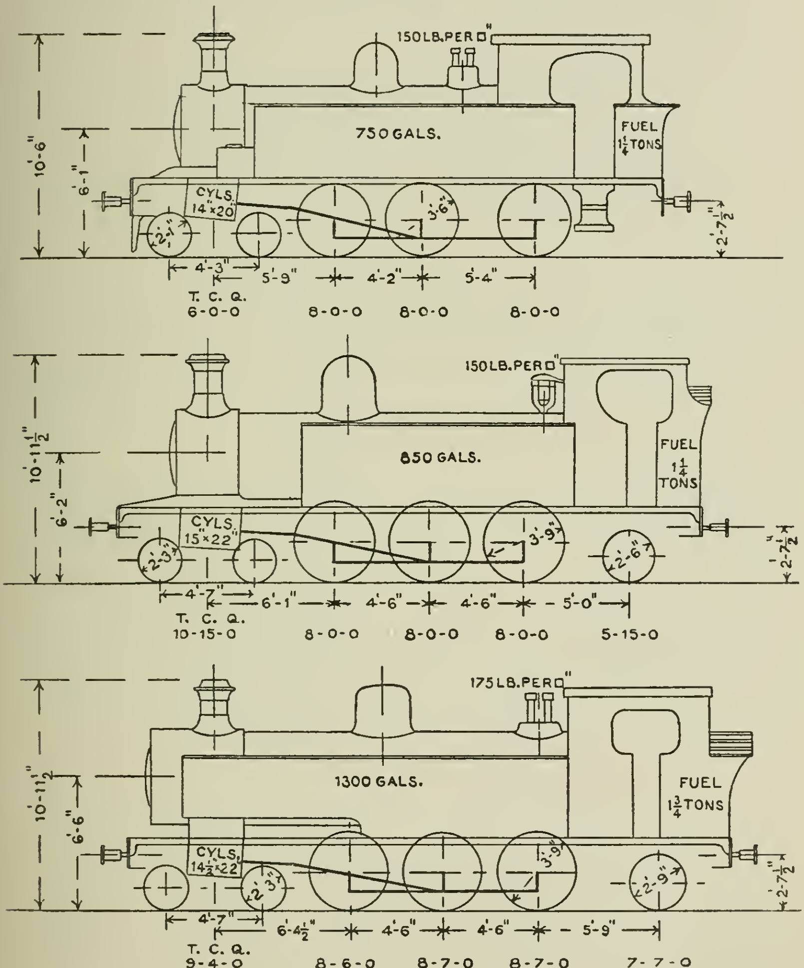
OPEN HIGH SIDED WAGON.




of any broad-gauge line. Fig. 9 and Plate 24 illustrate the standard six-compartment bogie-coach, and Fig. 10 a composite brake-van. The lavatory composites are fitted with lights at end, in order to afford a better view of the surrounding country; these are the only lavatory coaches in use on any Irish narrow-gauge railway. In the original six-wheel coaches, the extreme pairs of wheels have a lateral motion; these carriages are exceedingly comfortable in running, and go to show that a four-wheeled bogie is not a necessity even for sharp curves. The corridor observation

cars were introduced some years ago in order to afford tourists a larger and uninterrupted view of the scenery. The Donegal lines

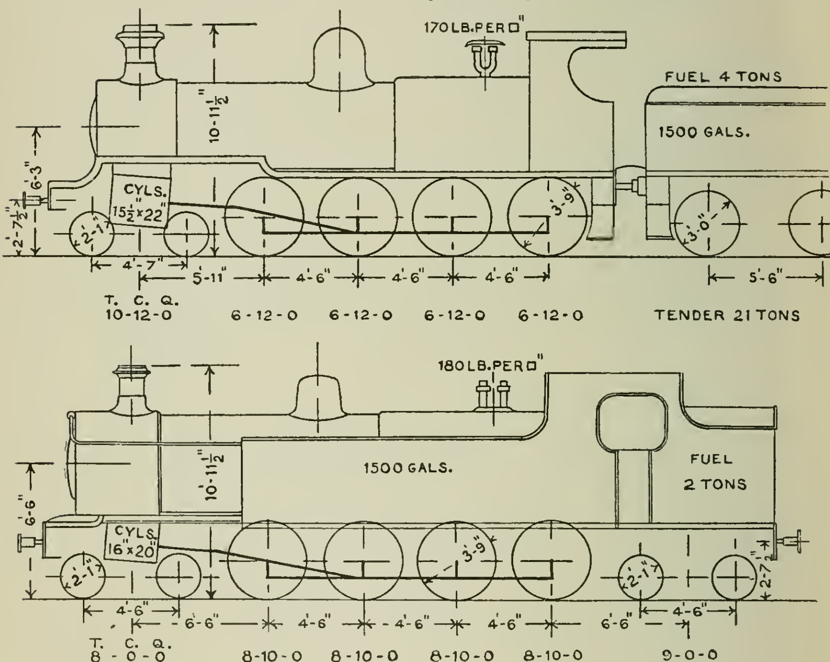
FIGS. 15, 16 AND 17.—*Londonderry and Lough Swilly Railway.*




were evidently pioneers in this respect, for, according to the Press, the London and North Western Railway only introduced them last year.

On Figs. 11 and 12 (page 612) are shown the latest types of combined covered goods and cattle and open goods wagons respectively. All the vehicles of the Irish narrow-gauge lines are equal to, if not larger in capacity, than those of similar type on any broad-gauge line.

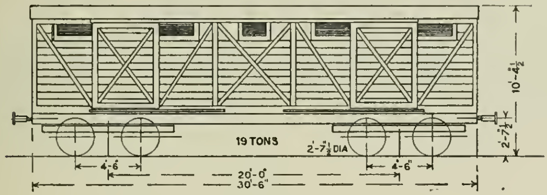
FIGS. 18 AND 19.—*Londonderry and Lough Swilly Railway.*




*Londonderry and Lough Swilly Railway.*—On Figs. 15 to 19 and Figs. 21 to 23, Plate 25, are illustrated the various types of engine in use on this railway; Fig. 15 is practically a duplicate of Fig. 3 on the Donegal line. Fig. 16 is a similar but more powerful engine, and Fig. 17 is a still heavier engine of the same type. Fig. 18 shows a radical departure from the earlier types, being a tender-engine, of which there are only two in use on Irish narrow-gauge lines. They possess great tractive power and are capable of very good work. Fig. 19 shows a new

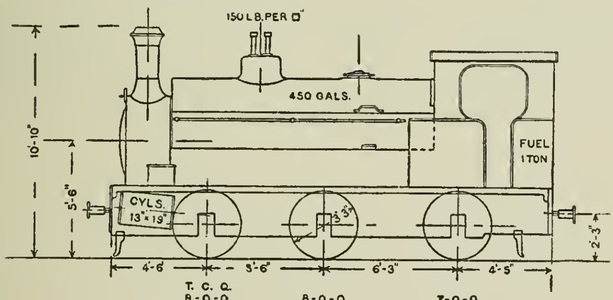
type of narrow-gauge locomotive, being a very powerful engine, and having a large tank capacity; it is now under construction, and consequently the author cannot give any particulars of its

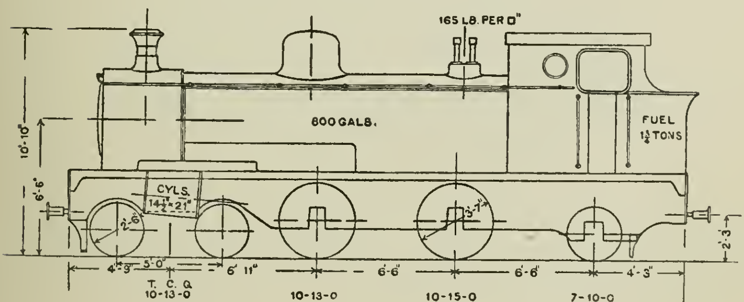
FIG. 20.—*Londonderry and Lough Swilly Railway.*




performance. A wagon of the Londonderry and Lough Swilly Company is illustrated in Fig. 20, and is similar to those in use on the County Donegal lines, but is lower, the buffer height being

FIGS. 24 AND 25.—*Ballycastle Railway.*



only 2 feet $7\frac{1}{2}$ inches, as compared with 2 feet $10\frac{1}{2}$ inches on the latter. This line possesses a type of covered goods wagon

FIGS. 26 AND 27.—*Ballycastle Railway.*

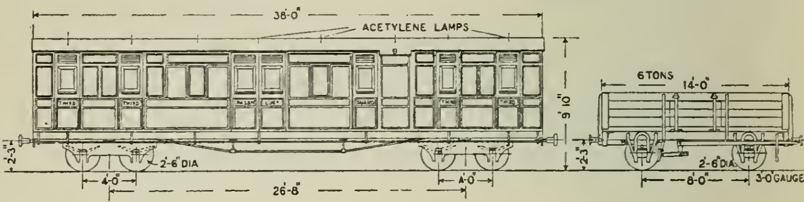



FIG. 30.—*Cork, Blackrock and Passage Railway.*

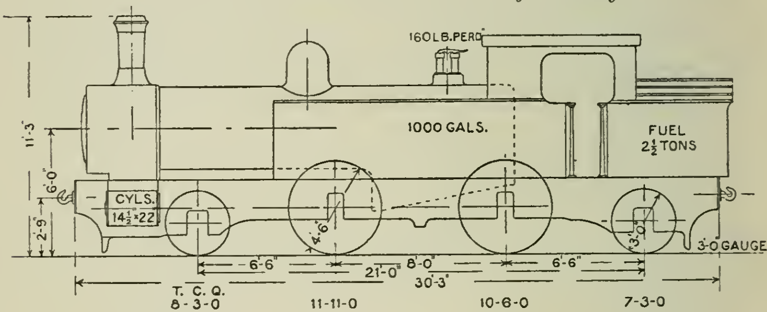
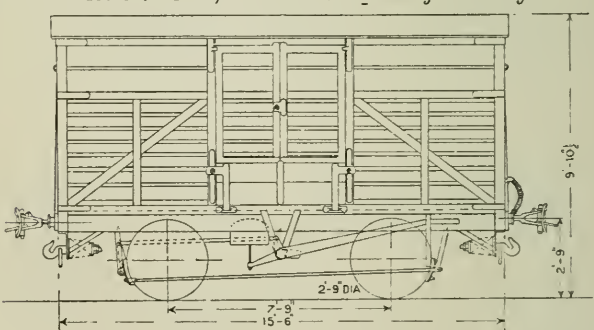



FIG. 32.—*Cork, Blackrock and Passage Railway.*

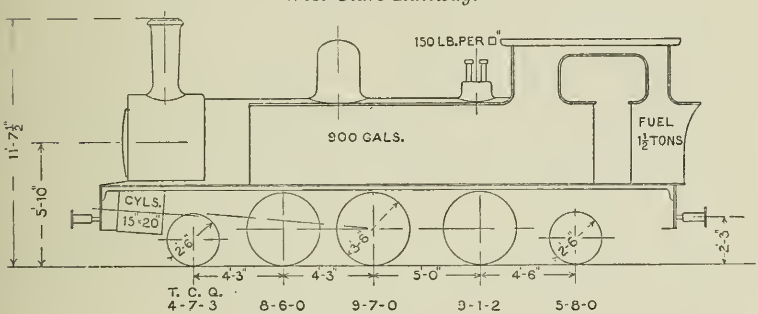



which is larger than that on any other narrow-gauge line, and has a carrying capacity of 19 tons.

*Ballycastle Railway.*—Figs. 24 and 25 (page 615) and Figs. 28 and 29, Plate 25, show the type of engine in use on this railway. It is the only line possessing a "saddle-tank." Fig. 25 shows an engine possessing about the same percentage of adhesive weight to total weight as Fig. 4 of the Donegal line, and which would be more efficient if another pair of coupled wheels were added. A carriage and wagon of the Ballycastle line are shown on Figs. 26 and 27. An eight-compartment carriage, accommodating eighty passengers, is unusual and confined to this line.

FIG. 33.

*West Clare Railway.*




*Cork, Blackrock and Passage Railway.*—In Fig. 30 and Fig. 31, Plate 25, are shown the only type of engine possessed by this line. It is an exceedingly handy machine, and evidently very efficient. It is the only 2–4–2 type in Ireland, excepting the Ballymena and Larne Railway, and has the largest coupled wheel of any narrow-gauge line in the Kingdom. Fig. 32 indicates the type of wagon in use; they differ from other narrow-gauge lines in having no side chains, but have the ordinary screw-coupling of the broad-gauge placed below the centre buffer and coupling.

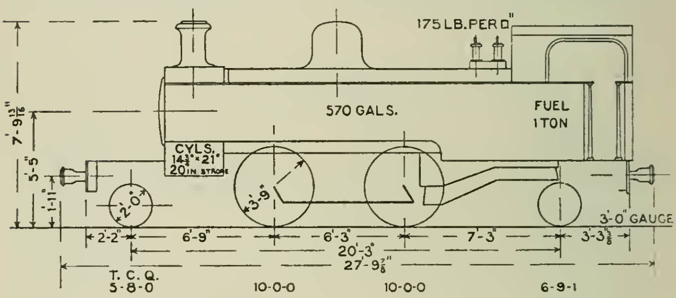
*West and South Clare Railways.*—Their latest type of engine is shown on Fig. 33 and Figs. 34 and 35, Plate 25. This is a powerful and very efficient machine.

*Schull and Skibbereen Railway.*—A type of engine is shown in Fig. 36, Plate 25.

*Midland Railway (N.C.C.): Ballymena and Larne Section.*—Figs. 37, 38, and 39, and Fig. 40, Plate 25, illustrate the type of engine and other vehicles used by this line. The latter engine

FIG. 37.

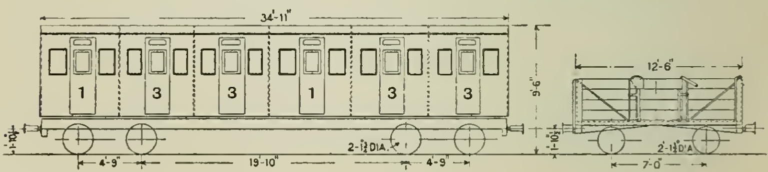
*Midland Railway (N.C.C.). Ballymena and Larne Section.*




is interesting as being the only narrow-gauge "compound" in Ireland, and is very efficient and economical; the carriages and wagons are smaller than, and differ from, most of those in use on other similar lines.

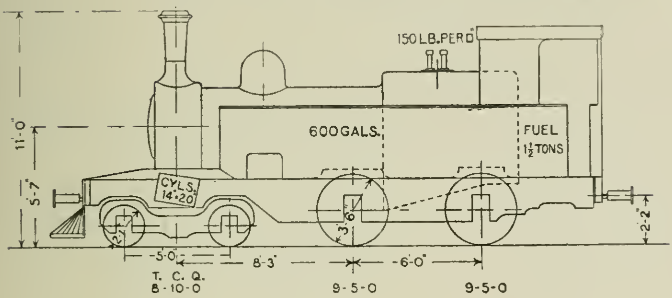
FIGS. 38 AND 39.

*Midland Railway (N.C.C.). Ballymena and Larne Railway.*




*Cavan and Leitrim Railway.*—Figs. 41 and 42, and Figs. 45 and 46, Plate 26, show the class of locomotive in use on this railway. They have exceptionally large fire-boxes, in order to burn Irish coal (Arigna), which is of very poor quality. The grates are fitted with

FIGS. 41 AND 42.—*Cavan and Leitrim Railway.*



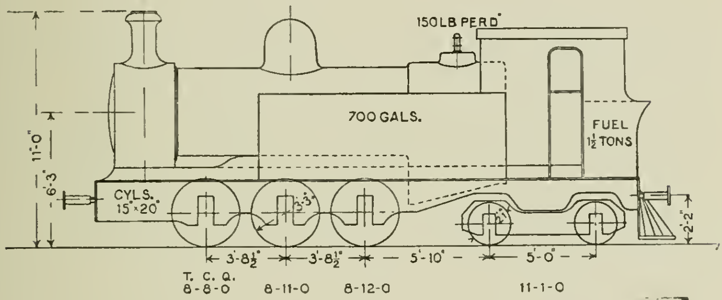




FIG. 43.—*Cavan and Leitrim Railway.*

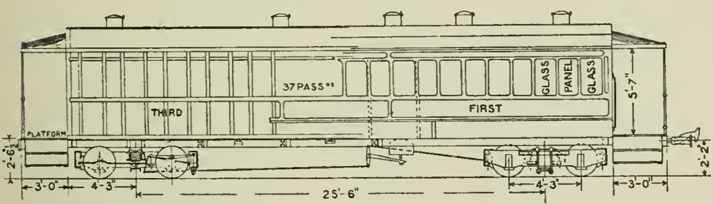
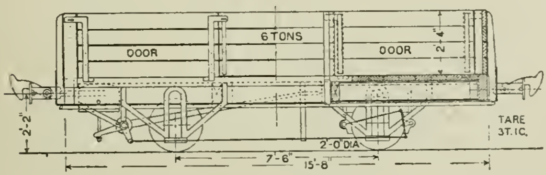
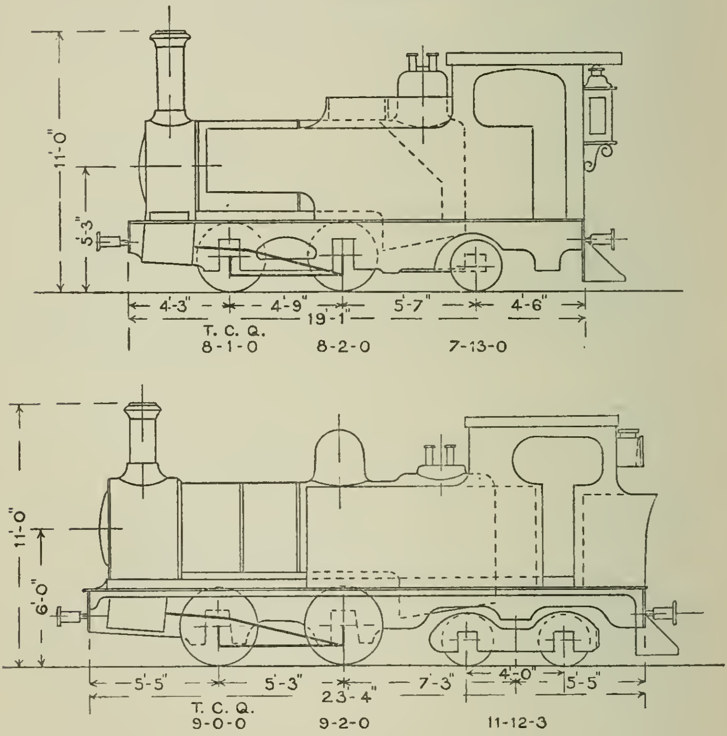



FIG. 44.—*Cavan and Leitrim Railway.*



drop bars at front end. These engines work partially on a tram-line, and generally bunker first; some of them are fitted so as to be operative from either end. The "cow-catcher" gives them a foreign appearance. The carriages and wagons are shown on Figs. 43 and 44, and Figs. 47 and 48, Plate 26. These vehicles differ considerably

FIGS. 49 AND 50.—*Clogher Valley Railway.*

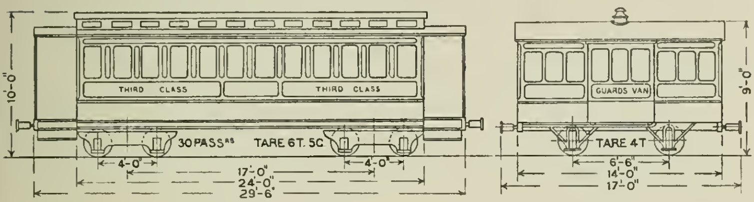



from those of the other narrow-gauge lines; they are very low, and the wagon stock is exceedingly wide.

*Clogher Valley Railway.*—Figs. 49 and 50 illustrate the engines of this line, which are of a type peculiar to this Company, and are relatively small. They generally work bunker first, like the Cavan

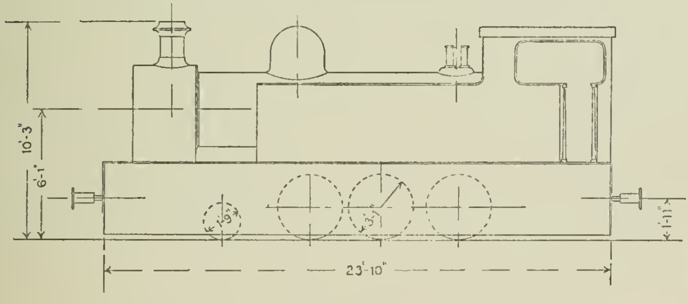
and Leitrim line, and are fitted with enormous head-lights illuminated by acetylene gas. Fig. 51 shows the type of corridor carriage in use; and Fig. 52 illustrates the brake van.

FIGS. 51 AND 52.—*Clogher Valley Railway.*




*Castlederg and Victoria Bridge Tramway.*—The small engine in use on this line is shown in Fig. 53. It does not call for special remark, except that the motion work is entirely enclosed. A passenger coach and wagon are shown on Figs. 54 and 55, Plate 26.

FIG. 53.—*Castlederg and Victoria Bridge Tramway.*




*Listowel and Ballybunion Railway.*—This line can hardly compare with any gauge of railway, as, being nominally a mono-rail, it has no gauge. It is interesting as a novelty in construction, and certainly it was inexpensive to build, having cost only £3,000 per mile, including land and rolling-stock. Each locomotive has three coupled wheels, 2 feet in diameter, and four guide-wheels, 10 inches in diameter. The two pairs of carrying wheels of the passenger vehicle are usually 19 inches in diameter, and two guide-wheels are

placed on each side. The carriages are 7 feet high, 8 feet 6 inches wide, and 18 feet long, and seat twelve passengers on each side. The usual maximum speed is 18 miles an hour; and they are said to be able to reach 30 miles. The locomotives pull 240 tons on the level, or 40 tons up 1 in 50. The curious construction of this line is shown by the photograph, Fig. 56, Plate 26. The line is constructed on what is known as the "Lartigue" system.

Two Tables are appended, Table 1 (pages 624–7) giving the principal dimensions of the various types of engine in use on the principal narrow-gauge lines in Ireland, with other information of interest. For comparison, Table 2 (pages 628–30) gives the working costs of some of the more important of the lines, as far as the locomotives, carriages and wagons are concerned, for the year 1911.

The author has to acknowledge, with thanks, the kind assistance rendered by the locomotive engineers of the following railways:—The Londonderry and Lough Swilly; Ballycastle; Cork, Blackrock and Passage; Ballymena and Larne; Cavan and Leitrim; Clogher Valley; West Clare; and Castlederg and Victoria Bridge. Also the following firms:—The North British Locomotive Co.; Messrs. Nasmyth, Wilson and Co.; The Metropolitan Amalgamated Carriage and Wagon Co.; Messrs. Hurst, Nelson and Co.; and Messrs. R. Y. Pickering and Co., for drawings and photographs. Also Messrs. Lawrence, of Dublin, for photos of the Listowel and Ballybunion system, and Mr. McCarthy, General Manager, for the other particulars of this interesting line.

The Paper is illustrated by Plates 23 to 26 and 37 Figs. in the letterpress.

# IRISH NARROW-GAUGE RAILWAY ROLLING-STOCK.

## TABLES 1 & 2.

TABLE 1 (*continued on opposite page*).

*Locomotives on Narrow-Gauge Railways in Ireland.*

*Table of Comparative Dimensions.*

| — | — | County Donegal Railways Joint Committee. Figs. 2 to 7 (pages 607-8). | | | | | |
|---|---|---|---|---|---|---|---|
| | | Class 1. | Class 2. | Class 3. | Class 4. | Class 5. | Class 5A. |
| 1 | Wheel arrangement | 2—4—0 | 4—6—0 | 4—4—4 | 4—6—4 | 2—6—4 | 2—6—4 |
| 2 | Diameter of cylinders . . . in. | 13 | 14 | 14 | 15 | 14 | 15½ |
| 3 | Stroke of cylinders . . . . in. | 20 | 20 | 20 | 21 | 21 | 21 |
| 4 | Working pressure lb. per. sq. in. | 120 | 150 | 150 | 160 | 175 | 160 |
| 5 | Heating surface, tubes . . sq. ft. | 510 | 547 | 637 | 643 | 637 | 646 |
| 6 | ,, ,, fire-box . sq. ft. | 45 | 57 | 76 | 80 | 76 | 78 |
| 7 | ,, ,, total . . sq. ft. | 555 | 604 | 713 | 723 | 713 | 724 |
| 8 | Grate area . . . . . . sq. ft. | 9·75 | 9·75 | 11·5 | 12 | 11·5 | 11·5 |
| 9 | Capacity of tanks . . . . gals. | 500 | 650 | 1,000 | 1,000 | 1,000 | 1,500 |
| 10 | ,, ,, bunkers . . . tons | 1 | 1 | 2½ | 1¾ | 2½ | 2½ |
| 11 | Weight on front bogie . t. c. q. | 4 0 0 | 6 9 3 | 7 8 0 | 8 15 0 | 6 15 0 | 8 2 1 |
| 12 | ,, ,, leading coupled wheels. t. c. q. | 8 0 0 | 6 17 2 | — | 8 12 0 | 8 15 0 | 10 2 1 |
| 13 | ,, ,, driving coupled wheels. t. c. q. | 8 0 0 | 8 11 1 | 10 13 0 | 8 16 0 | 9 0 0 | 10 6 0 |
| 14 | ,, ,, trailing coupled wheels. t. c. q. | — | 8 14 1 | 10 3 2 | 8 12 0 | 8 15 0 | 10 4 0 |
| 15 | ,, ,, rear bogie . t. c. q. | — | — | 10 6 0 | 9 15 0 | 10 5 0 | 11 14 0 |
| 16 | Total weight in working order . . t. c. q. | 20 0 0 | 30 12 3 | 38 10 3 | 44 10 0 | 43 10 0 | 50 8 2 |
| 17 | ,, ,, on coupled wheels . t. c. q. | 16 0 0 | 24 3 0 | 20 16 3 | 26 0 0 | 27 10 0 | 30 12 1 |
| 18 | Tractive force per lb. of mean pressure in cylinders . . . lb. | 80·4 | 93·3 | 81·7 | 105 | 85·75 | 105·10 |
| 19 | Tractive force at 75 per cent. of boiler pressure . . . . . lb. | 7,236 | 10,496 | 9,191 | 12,600 | 11,233 | 12,612 |
| 20 | Adhesion at 500 lb. per ton . . lb. | 8,000 | 12,075 | 10,418 | 13,000 | 13,750 | 15,306 |
| 21 | Ratio of adhesive to total weight . per cent. | 80 | 78·8 | 54 | 58·4 | 63·2 | 61 |
| 22 | Diameter of coupled wheels ft. in. | 3 6 | 3 6 | 4 0 | 3 9 | 4 0 | 4 0 |
| 23 | Rigid wheel-base . . . . ft. in. | 6 0 | 9 0 | 6 0 | 9 0 | 10 0 | 10 0 |
| 24 | Total ,, ,, . . . . . ft. in. | 11 6 | 16 7½ | 26 1½ | 25 2 | 26 3 | 27 3 |
| 25 | Radius of sharpest curve engine will negotiate . . . . . ft. | 264 | 300 | 350 | 350 | 350 | 350 |
| 26 | Steepest gradient over which engine works . . . . . . . . . | 1 in 40 | 1 in 40 | 1 in 40 | 1 in 40 | 1 in 40 | 1 in 40 |
| 27 | Load taken on same . . . tons | 80 | 100 | 90 | 130 | 140 | 170 |
| 28 | Weight of rail used . lb. per yd. | 50 & 60 | 50 & 60 | 50 & 60 | 50 & 60 | 50 & 60 | 50 & 60 |
| 29 | Type of brake in use . . . . . | Vacuum | Vacuum | Vacuum | Vacuum | Vacuum | Vacuum |

(*continued on next page*) TABLE 1.

*Locomotives on Narrow-Gauge Railways in Ireland.*

*Table of Comparative Dimensions.*

| Londonderry and Lough Swilly Railway. Figs. 15 to 19 (pages 613–4). | | | | | Ballycastle Railway. Figs. 24 and 25 (page 615). | | — |
|---|---|---|---|---|---|---|---|
| — | — | — | Tender Engine. | — | Saddle-Tank. | — | |
| 4—6—0 | 4—6—2 | 4—6—2 | 4—8—0 | 4—8—4 | 0—6—0 | 4—4—2 | 1 |
| 14 | 15 | 14½ | 15½ | 16 | 13 | 14½ | 2 |
| 20 | 22 | 22 | 22 | 20 | 19 | 21 | 3 |
| 150 | 150 | 175 | 170 | 180 | 150 | 165 | 4 |
| 565 | 700·75 | 728 | 889 | 871·3 | 531 | 769 | 5 |
| 63 | 76·5 | 75 | 115·9 | 132·2 | 51 | 83 | 6 |
| 628 | 777·25 | 903 | 1004·9 | 1003·5 | 582 | 852 | 7 |
| 9·5 | 12·5 | 11·5 | 15 | 17 | 7 | 12 | 8 |
| 750 | 850 | 1,300 | 1,500 | 1,500 | 450 | 800 | 9 |
| 1¼ | 1¼ | 1¾ | 5 | 2½ | 1 | 1¾ | 10 |
| 6 0 0 | 10 15 0 | 9 4 0 | 10 12 0 | 8 0 0 | — | 10 3 0 | 11 |
| 8 0 0 | 8 0 0 | 8 6 0 | — | — | 9 0 0 | 10 13 0 | 12 |
| 8 0 0 | 8 0 0 | 8 7 0 | 26 8 0 | 34 0 0 | 8 0 0 | 10 15 0 | 13 |
| 8 0 0 | 8 0 0 | 8 7 0 | — | — | 7 0 0 | — | 14 |
| — | 5 15 0 | 7 7 0 | — | 9 0 0 | — | 7 10 0 | 15 |
| 30 0 0 | 40 10 0 | 41 11 0 | 58 0 0 | 51 0 0 | 24 0 0 | 39 1 0 | 16 |
| 24 0 0 | 24 0 0 | 25 0 0 | 26 8 0 | 34 0 0 | 24 0 0 | 21 8 0 | 17 |
| 93·3 | 110 | 102·78 | 117·45 | 113·77 | 82·33 | 99·13 | 18 |
| 10,496 | 12,375 | 13,489 | 14,974 | 15,358 | 9,262 | 12,267 | 19 |
| 12,000 | 12,000 | 12,500 | 13,200 | 17,000 | 12,000 | 10,700 | 20 |
| 80 | 59·2 | 60·1 | 45 | 66·6 | 100 | 54 | 21 |
| 3 6 | 3 9 | 3 9 | 3 9 | 3 9 | 3 3 | 3 7 | 22 |
| 9 6 | 9 0 | 9 0 | 13 6 | 13 6 | 11 9 | 6 6 | 23 |
| 17 4½ | 22 4½ | 23 6 | 40 5 | 31 0 | 11 9 | 22 5 | 24 |
| 300 | 300 | 400 | 400 | 400 | 260 | 330 | 25 |
| 1 in 50 | 1 in 50 | 1 in 50 | 1 in 50 | 1 in 50 | 1 in 48 | 1 in 48 | 26 |
| 120 | 130 | 130 | 150 | 150 | 128 | 181 | 27 |
| 50 | 50 | 50 | 38–50 | 50 | 45 | 45 | 28 |
| Vacuum | Vacuum | Vacuum | Vacuum | Vacuum | Steam | Vacuum and Steam | 29 |

TABLE 1 (*continued from previous page*).

*Locomotives on Narrow-Gauge Railways in Ireland.*

*Table of Comparative Dimensions.*

| — | — | Cork, Blackrock and Passage. Fig. 30. | West Clare Railway. Fig. 33. | Ballymena and Larne. N. C. C. Fig. 37. |
|---|---|---|---|---|
| 1 | Wheel arrangement . . . . . | 2—4—2 | 2—6—2 | 2—4—2 |
| 2 | Diameter of cylinders . . . in. | 14½ | 15 | 14¾ and 21 |
| 3 | Stroke of cylinders . . . . in. | 22 | 20 | 20 |
| 4 | Working pressure lb. per sq. in. | 160 | 150 | 175 |
| 5 | Heating surface, tubes . . sq. ft. | 721 | 666·57 | 614·83 |
| 6 | ,, ,, fire-box . sq. ft. | 80 | 77 | 63 |
| 7 | ,, ,, total . . sq. ft. | 801 | 743·57 | 677·83 |
| 8 | Grate area . . . . . . . sq. ft. | 12 | 11·18 | 11·29 |
| 9 | Capacity of tanks . . . gals. | 1,000 | 900 | 570 |
| 10 | ,, ,, bunkers . . . tons | 2½ | 1½ | 1 |
| 11 | Weight on front bogie . t. c. q. | 8 3 0 | 4 7 3 | 5 8 1 |
| 12 | ,, ,, leading coupled wheels. t. c. q. | — | 8 6 0 | 10 0 0 |
| 13 | ,, ,, driving coupled wheels. t. c. q. | 11 11 0 | 9 7 0 | 10 0 0 |
| 14 | ,, ,, trailing coupled wheels. t. c. q. | 10 6 0 | 9 1 2 | — |
| 15 | ,, ,, rear bogie . t. c. q. | 7 3 0 | 5 8 0 | 6 9 1 |
| 16 | Total weight in working order . t. c. q. | 37 3 0 | 36 10 1 | 31 17 2 |
| 17 | ,, ,, on coupled wheels . t. c. q. | 21 17 0 | 27 14 2 | 20 0 0 |
| 18 | Tractive force per lb. of mean pressure in cylinders . . . lb. | 85·65 | 107·14 | 105·94 |
| 19 | Tractive force at 75 per cent. of boiler pressure . . . . . lb. | 10,278 | 12,053 | 13,904 |
| 20 | Adhesion at 500 lb. per ton . . lb. | 10,925 | 13,862 | 10,000 |
| 21 | Ratio of adhesive to total weight . per cent. | 58 | 75·8 | 62·7 |
| 22 | Diameter of coupled wheels ft. in. | 4 6 | 3 6 | 3 9 |
| 23 | Rigid wheel-base . . . . ft. in. | 8 0 | 9 3 | 6 3 |
| 24 | Total ,, ,, . . . . ft. in. | 21 0 | 18 0 | 20 3 |
| 25 | Radius of sharpest curve engine will negotiate . . . . . ft. | 660 | 990 | 330 |
| 26 | Steepest gradient over which engine works . . . . . . . . . | 1 in 64 | 1 in 60 | 1 in 33 |
| 27 | Load taken on same . . . tons | 150 | 150 | 100 |
| 28 | Weight of rail used . lb. per yd. | 62 | — | 50 & 65 |
| 29 | Type of brake in use . . . . . | Vacuum | Smith's Vacuum | Vacuum |

(*concluded from opposite page*) TABLE 1.

*Locomotives on Narrow-Gauge Railways in Ireland.*

*Table of Comparative Dimensions.*

| Cavan and Leitrim. | | Castlederg and Victoria Bridge Tramway. | | Clogher Valley Railway. | | — |
|---|---|---|---|---|---|---|
| Fig. 41. | Fig. 42. | — | Fig. 53. | Fig. 49. | Fig. 50. | |
| 4—4—0 | 0—6—4 | 0—4—0 | 2—6—0 | 0—4—2 | 0—4—4 | 1 |
| 14 | 15 | 12 | 13½ | 13½ | 14 | 2 |
| 20 | 20 | 15 | 18 | 18 | 20 | 3 |
| 150 | 150 | 150 | 160 | 140 | 160 | 4 |
| 466 | 680·75 | 235·8 | 542·4 | 466 | 548 | 5 |
| 67·5 | 68·75 | 33·5 | 54 | 48 | 57 | 6 |
| 533·5 | 749·5 | 269·3 | 596·4 | 514 | 605 | 7 |
| 11·5 | 14 | 7·3 | 9·25 | 10 | 12·25 | 8 |
| 600 | 700 | 450 | 600 | 600 | 700 | 9 |
| 1 | 1½ | ½ | ¾ | 1 | 1¼ | 10 |
| 8 10 0 | — | — | 5 0 0 | — | — | 11 |
| — | 8 8 0 | 9 0 0 | 7 0 0 | 8 1 0 | 9 0 0 | 12 |
| 9 5 0 | 8 11 0 | 9 0 0 | 7 0 0 | 8 2 0 | 9 2 0 | 13 |
| 9 5 0 | 8 12 0 | — | 7 0 0 | — | — | 14 |
| — | 11 1 0 | — | — | 7 13 0 | 11 12 3 | 15 |
| 27 0 0 | 36 12 0 | 18 0 0 | 26 0 0 | 23 16 0 | 29 14 3 | 16 |
| 18 10 0 | 25 11 0 | 18 0 0 | 21 0 0 | 16 3 0 | 18 2 0 | 17 |
| 93·33 | 115·38 | 65·44 | 88·66 | 91·12 | 98 | 18 |
| 10,499 | 12,980 | 7,362 | 10,639 | 9,567 | 11,760 | 19 |
| 9,250 | 12,775 | 9,000 | 10,500 | 8,075 | 9,050 | 20 |
| 68·5 | 69·9 | 100 | 80·7 | 67·3 | 60·8 | 21 |
| 3 6 | 3 3 | 2 9 | 3 1 | 3 0 | 3 4 | 22 |
| 6 0 | 7 5 | 6 6 | 7 0 | 4 9 | 5 3 | 23 |
| 19 3 | 18 3 | 6 6 | 11 2 | 10 2 | 14 6 | 24 |
| 240 | 240 | 120 | 120 | 110 | 110 | 25 |
| 1 in 30 | 1 in 30 | 1 in 30 | 1 in 30 | 1 in 30 | 1 in 30 | 26 |
| 80 | 110 | 50 | 74 | 70 | 90 | 27 |
| 45 | 45 | 30 | 30 | 45 | 45 | 28 |
| Westinghouse | Westinghouse | Westinghouse | Westinghouse | Vacuum | Vacuum | 29 |

TABLE 2 (*continued on opposite page*).

*Irish Narrow-Gauge Rolling-Stock. Table of Working Costs for* 1911.

| — | Item. | A | B | C |
|---|---|---|---|---|
| | *Locomotives.* | | | |
| 1 | Running wages per engine-mile . . d. | 1·80 | 1·28 | 1·56 |
| 2 | ,, ,, ,, train-mile . . ,, | 2·17 | 1·48 | 1·74 |
| 3 | Coal, cost per engine-mile . . . ,, | 3·69 | 4·24 | 3·07 |
| 4 | ,, ,, ,, train-mile . . . ,, | 4·45 | 4·48 | 3·43 |
| 5 | Water, cost per engine-mile . . ,, | 0·15 | 0·09 | — |
| 6 | ,, ,, ,, train-mile . . . ,, | 0·18 | 0·11 | — |
| 7 | Stores, cost per engine-mile . . ,, | 0·24 | 0·27 | 0·38 |
| 8 | ,, ,, ,, train-mile . . . ,, | 0·30 | 0·31 | 0·42 |
| 9 | Total running costs per engine-mile . ,, | 5·90 | 5·88 | 5·01 |
| 10 | ,, ,, ,, ,, train-mile . ,, | 7·10 | 6·78 | 5·59 |
| 11 | Repairs, costs per engine-mile . . ,, | 1·39 | 1·73 | 2·34 |
| 12 | ,, ,, ,, train-mile . . ,, | 1·69 | 1·98 | 2·61 |
| 13 | Superintendence, costs per engine-mile ,, | 0·13 | 0·18 | 0·29 |
| 14 | ,, ,, ,, train-mile . ,, | 0·16 | 0·20 | 0·32 |
| 15 | Total locomotive costs per engine-mile ,, | 7·42 | 7·79 | 7·64 |
| 16 | ,, ,, ,, ,, train-mile . ,, | 8·95 | 8·96 | 8·53 |
| 17 | Cost of repairs per engine per annum £ s. d. | 122 7 9 | 154 0 8 | 123 2 11 |
| 18 | Engine-miles run per engine per annum . | 21,046 | 21,466 | 12,609 |
| 19 | Train-miles ,, ,, ,, ,, ,, . | 17,565 | 18,616 | 11,302 |
| 20 | Average miles run per engine between repairs | 69,190 | 30,000 | 21,000 |
| 21 | Coal burnt per engine-mile . . lb. | 36 | 44·4 | 32·86 |
| 22 | ,, ,, ,, train-mile . . . ,, | 43·5 | 51·19 | 36·65 |
| 23 | Number per mile of line . . . . | 0·16 | 0·17 | 0·24 |
| | *Carriages.* | | | |
| 24 | Costs of repairs per train-mile . . d. | 0·70 | 0·78 | 0·94 |
| 25 | ,, ,, ,, ,, carriage per annum £ s. d. | 14 5 5 | 17 6 4 | 12 3 4 |
| 26 | Number per mile of line . . . . | 0·44 | 0·54 | 0·92 |
| | *Wagons.* | | | |
| 27 | Cost of repairs per train-mile . . d. | 0·61 | 0·36 | 0·76 |
| 28 | ,, ,, ,, ,, wagon per annum £ s. d. | 2 4 2 | 1 5 9 | 2 8 9 |
| 29 | Number per mile of line . . . . | 2·5 | 2·8 | 3·63 |

(*continued on next page*) TABLE 2.

*Irish Narrow-Gauge Rolling-Stock. Table of Working Costs for* 1911.

| D | E | F | G | H | J | — |
|---|---|---|---|---|---|---|
| 1·34 | 2·15 | 1·90 | 2·26 | 1·80 | — | 1 |
| 1·80 | 2·18 | 2·36 | 2·42 | 1·98 | 1·23 | 2 |
| 2·47 | 4·07 | 2·63 | 3·71 | 2·94 | — | 3 |
| 3·33 | 4·11 | 3·25 | 3·97 | 3·21 | 4·07 | 4 |
| 0·10 | 0·08 | 0·09 | 0·13 | 0·09 | — | 5 |
| 0·13 | 0·09 | 0·11 | 0·14 | 0·10 | — | 6 |
| 0·18 | 0·32 | 0·28 | 0·23 | 0·23 | — | 7 |
| 0·25 | 0·33 | 0·34 | 0·25 | 0·26 | 0·44 | 8 |
| 4·09 | 6·62 | 4·90 | 6·32 | 5·06 | — | 9 |
| 5·51 | 6·71 | 6·06 | 6·78 | 5·55 | 5·74 | 10 |
| 0·81 | 1·86 | 1·26 | 2·57 | 1·96 | — | 11 |
| 1·10 | 1·89 | 1·56 | 2·75 | 2·15 | 2·74 | 12 |
| 0·17 | 0·34 | 0·13 | 0·28 | 0·32 | — | 13 |
| 0·23 | 0·35 | 0·15 | 0·30 | 0·35 | — | 14 |
| 5·07 | 8·82 | 6·29 | 9·30 | 7·34 | — | 15 |
| 6·84 | 8·95 | 7·77 | 9·97 | 8·05 | 8·48 | 16 |
| 116 1 5 | 106 19 4 | 110 19 1 | 128 9 0 | 122 4 4 | 81 15 2 | 17 |
| 34,485 | 14,843 | 22,942 | 11,980 | 14,994 | — | 18 |
| 25,560 | 14,660 | 18,530 | 11,187 | 13,672 | 7,135 | 19 |
| 97,000 | — | 41,185 | 41,905 | 50,000 | 18,500 | 20 |
| 23·53 | 33·12 | 29·41 | 38 | 21·67 | — | 21 |
| 31·71 | 33·46 | 36·58 | 40·69 | 23·77 | 24 | 22 |
| 0·25 | 0·20 | 0·22 | 0·18 | 0·18 | 0·41 | 23 |
| 0·94 | 1·14 | 0·72 | 0·64 | 0·65 | 1·46 | 24 |
| 12 6 10 | 20 14 2 | 18 11 5 | 11 12 1 | 14 8 10 | 3 14 6 | 25 |
| 1·75 | 0·67 | 0·68 | 0·49 | 0·54 | 0·96 | 26 |
| 0·23 | 1·03 | 1·88 | 2·03 | 0·86 | 1·46 | 27 |
| 3 3 2 | 4 13 2 | 2 12 4 | 5 11 8 | 3 3 3 | 3 14 6 | 28 |
| 1·75 | 2·62 | 12·68 | 2·94 | 2·89 | 3·72 | 29 |

TABLE 2 (*concluded from page* 628).

*Irish Narrow-Gauge Rolling-Stock. Table of Working Costs for* 1911.

| Item. | K | L | Average of Narrow-Gauge Lines. | Average of Irish Broad-Gauge Lines. |
|---|---|---|---|---|
| *Locomotives.* | | | | |
| Running wages per engine-mile . . *d.* | — | — | 1·76 | — |
| ,, ,, ,, train-mile . . ,, | 2·02 | 2·95 | 2·03 | 2·62 |
| Coal, cost per engine-mile . . . ,, | — | — | 3·35 | — |
| ,, ,, ,, train-mile . . . ,, | 3·64 | 4·39 | 3·84 | 4·33 |
| Water, cost per engine-mile . . ,, | — | — | 0·10 | — |
| ,, ,, ,, train-mile . . . . ,, | 0·05 | 0·07 | 0·10 | — |
| Stores, cost per engine-mile . . ,, | — | — | 0·26 | — |
| ,, ,, ,, train-mile . . . . ,, | 0·37 | 0·37 | 0·33 | 0·55 |
| Total running costs per engine-mile . ,, | — | — | 5·47 | — |
| ,, ,, ,, ,, train-mile . ,, | 6·08 | 7·78 | 6·33 | 7·58 |
| Repairs, costs per engine-mile . . ,, | — | — | 1·74 | — |
| ,, ,, ,, train-mile . . ,, | 0·91 | 3·93 | 2·11 | 2·54 |
| Superintendence, costs per engine-mile ,, | — | — | 0·23 | — |
| ,, ,, ,, train-mile . ,, | 0·14 | 0·40 | 0·26 | 0·15 |
| Total locomotive costs per engine-mile ,, | — | — | 7·45 | — |
| ,, ,, ,, ,, train-mile . ,, | 7·12 | 12·11 | 8·70 | 10·29 |
| Cost of repairs per engine per annum £ *s. d.* | 47 5 2 | 152 5 2 | 115 0 10 | — |
| Engine-miles run per engine per annum . | — | — | 19,298 | — |
| Train-miles ,, ,, ,, ,, ,, . | 12,127 | 9,284 | 14,512 | 10,763 |
| Average miles run per engine between repairs | — | — | 46,097 | — |
| Coal burnt per engine-mile . . lb. | — | — | 32·32 | — |
| ,, ,, ,, train-mile . . . . ,, | — | — | 35·67 | — |
| Number per mile of line . . . . | 0·33 | 0·21 | 0·23 | 0·32 |
| *Carriages.* | | | | |
| Costs of repairs per train-mile . . *d.* | 1·12 | 1·08 | 0·92 | 5·23 |
| ,, ,, ,, ,, carriage per annum £ *s. d.* | 12 13 0 | 16 15 0 | 14 0 11 | 16 19 7 |
| Number per mile of line . . . . | 1·50 | 0·53 | 0·77 | 1·25 |
| *Wagons.* | | | | |
| Cost of repairs per train-mile . . *d.* | 0·64 | 1·10 | 0·99 | 3·53 |
| ,, ,, ,, ,, wagon per annum £ *s. d.* | 3 16 0 | 4 7 5 | 3 7 3 | 2 16 7 |
| Number per mile of line . . . . | 3·16 | 2·08 | 3·60 | 6·78 |

*Discussion.*

On the motion of the President, a hearty vote of thanks was accorded to the author for his interesting and instructive Paper.

Mr. James E. Darbishire, in opening the discussion, said he had listened to the Paper dealing with the narrow-gauge railways of Ireland with very great pleasure, because he was concerned at the very beginning with the introduction of the narrow-gauge system into the country. He looked upon himself as godfather to the locomotive "Alice" shown in Fig. 13, Plate 23. This was one of three engines originally built for the West Donegal Railway, which was now part of the County Donegal Railways. They were called after three young ladies; they all turned out to be very useful engines, and from the Paper he learnt that one of them was still in service.

When the narrow-gauge railways of Ireland were first mooted, there was a great deal of discussion amongst locomotive engineers as to the proper system of locomotive to introduce upon them. He thought that a good many mistakes were made at the beginning, but nevertheless some very useful engines were put upon the Ballymena and Larne Railway of the type originally built for the Isle of Man Railways by Beyer, Peacock and Co. Those engines were found to answer very well, and the same design with modifications was subsequently followed out on the further railways that were made in Ireland. Both the Ballymena and Larne Railway and the Ballymena, Cushendall and Red Bay Railway suffered at first from the fact that the rails were too light, but the road had been very much strengthened. When the West Donegal line was commenced, a discussion ensued as to the type of engine that should be used, and the little engine to which he had already referred, "Alice," in Fig. 13, was designed with a radial axle-box in front and four wheels coupled and a wide fire-box, which was put into frames outside the wheels. That little engine did very well indeed. Later on his firm were consulted with regard to the

(Mr. James E. Darbishire.)
engines for the Clogher Valley line. He did not know whether the original Clogher Valley engine, which was more or less of a tramway engine, was shown among the illustrations. The conditions in that case were quite different from those existing in the other lines. The Clogher Valley Railway ran very largely along the high road, and it was not so much of a railway as the others.

He (Mr. Darbishire) came over to Ireland several times for the purpose of considering the question of the engines, and was very much interested in the Irish narrow-gauge lines. Subsequently he went out to South America, and saw a great deal of narrow-gauge railway construction in the hilly countries of Brazil. In that case the lines were again spoiled by building them too cheaply. £4,000 per mile was supposed to be the cost of the line, including the rolling-stock. He did not mean to say that that was the actual cost, but that was the figure that was talked about. The lines were laid with 40-lb. rails, and numerous difficulties were experienced with the locomotives. In that case they also had to adopt the principle that was subsequently put in force in Ireland of substituting stronger rails.

The remark made by the author on page 599, as to the principal saving in the construction of narrow-gauge railways over broad-gauge lines being the cost of the land, was true to some extent, but that was not altogether so in Ireland, because a good many of the broad-gauge lines were altered to narrow-gauge for convenience of working. In that case the cost of the land was not a matter for consideration, because the narrow-gauge line was laid upon the broad-gauge formation. It must also be borne in mind that the gauge of the ordinary railways in Ireland was exceptionally broad, 5 feet 3 inches as against 4 feet 8½ inches in England, so that the comparison between the broad and the narrow-gauge in Ireland was rather more favourable to the narrow-gauge than it was in England.

Mr. John A. F. Aspinall (Past-President) said the author made the following remark (page 599): "No railway should be built of narrow gauge if the cost will exceed £5,000 per mile." He did

not think that any narrow-gauge railway in Ireland had been made for less than about £6,000 a mile, and the author himself gave particulars of one—which he presumed was in Ireland—which cost £11,500 a mile. In England he did not think any narrow-gauge railway had been made for less than £8,000 a mile, largely due to the fact that the requirements for the construction of narrow-gauge railways were exceedingly onerous. If the members wished to see a country in which narrow-gauge railways were constructed with great economy and with small capital cost, they should go to Belgium. In that country there were something like 2,500 miles of narrow-gauge railway carefully laid out, arranged to act as feeders to the main lines. They ran through numerous small villages, and were so constructed that there were no fences, no bridges, and no signalling, with the result that the average cost of the whole of the narrow-gauge railways of Belgium had worked out at about £3,400 a mile. A most interesting return in connection with these railways had been published by the Belgian Government, which showed in most minute detail what they had cost for land, for rolling-stock, for building, and for every single item, while the cost of the operation was also given. Not only in England, but in Ireland also, too many difficulties had been put in the way of cheap construction by the requirements of the Board of Trade. It was quite true that the Board of Trade had recently issued a new Bill in connection with light railways, which he presumed might be passed into law, but it was a timid and a halting measure, and did not even carry out the suggestions which were made quite recently by their senior inspector, Colonel Yorke, who reported upon the subject of light railways with the object of seeing whether it was necessary to have some amendments made in the rules which regulated the Board of Trade in passing such railways before they were used by the public. He thought that was to be regretted, because in England, where there was such an immense number of manufactures both small and great in the villages, it would be of the greatest possible advantage to have those somewhat slow-running but cheap railways for carrying passengers and material to the larger lines at low price.

(Mr. John A. F. Aspinall.)

The information which the author had given was very interesting, and no doubt he could show that very good results had been obtained on the Irish narrow-gauge railways. But what could be done upon a narrow-gauge railway was best illustrated by what had been done on the South African lines. Those lines had a 3-foot 6-inch gauge with very wide rolling-stock, comprising sleeping-cars and dining-cars, and very powerful locomotives hauling trains of considerable tonnage, and there was little doubt that if a narrow-gauge railway was properly constructed, an enormous amount of work could be done by it. He quite agreed that considerable hesitation should be shown before putting down a narrow-gauge line in a country where broad-gauge railways existed, unless some cheap method existed of transfer from one to the other; but there were circumstances in which a narrow-gauge line was really advantageous and sometimes absolutely necessary. He doubted himself whether it would be advantageous to Ireland to extend the system for any more miles than were actually working at the present time. He would prefer to see any extensions of the Irish Railways made on the standard 5-foot 3-inch gauge.

Mr. BOWMAN MALCOLM said he had been connected with the narrow-gauge railways of Ireland for a great number of years, going back to the time when the West Donegal Railway was constructed. At the present time the first two narrow-gauge lines built in Ireland were part of the Northern Counties Committee's system. The line from Ballymena to Parkmore was built originally (1875) as a mineral line, and had three small saddle-tank engines, two of which were in existence at the present time, although only used for supernumerary purposes. Shortly afterwards the line running from Larne to Ballymena across country was made (1878), and rolling-stock of a peculiar type had to be designed for it owing to the very light rails which were originally used. The rails were 40 lb. to the yard, and the loads were kept down to 8 tons per axle. If reference was made to Table 1 (page 626), it would be observed that the heaviest engines of the Northern Counties narrow-gauge system weighed only 31 tons 17 cwt., which was very low compared

with the weights of the majority of the other engines on the principal lines mentioned. That was partly due no doubt to the small tanks, which only contained about 600 gallons, whereas some of the other engines carried as much as 1,000 and 1,500 gallons. He always looked upon the gross weight of the engine as a very important factor on a system such as the Ballymena and Larne, because the line possessed long gradients of 1 in 37 and some short lengths of 1 in 33. The members would notice that the engine specified on page 626 and Fig. 37 (page 618) would take a gross load of 100 tons up the latter gradient, which was, he thought, very good work. The trouble they suffered from on that particular gradient was that not only was it very severe, but it was on a series of 5-chain curves, the gradient being uncompensated. It might be a matter of interest to the members if he stated that the engine shown in Fig. 37 (page 618) was a two-cylinder compound. The company had a number of those engines, and it was now their standard.

A mistake was made very early in the building of narrow-gauge railways in Ireland in that no standard was laid down for the height of the centre of the buffer from the rail, and it might astonish the members to learn that that difference amounted to 11½ inches, so that stock on one narrow-gauge line could not be transferred to another. There was a very considerable difference in the height of the Northern Counties Committee's stock, the stock of the Ballycastle Company, which was in close physical connection with the Northern Counties' system, and of the County Donegal lines generally; so that assuming the lines were amalgamated they would not be able to transfer the stock from one line to another without considerable alteration. As a matter of fact, a rather ingenious arrangement had been made for the purpose of working the County Donegal Joint Committee's wagon stock over the Londonderry and Lough Swilly Railway. In that case there was a difference of 3 inches. A two-storey buffer had been constructed with a long head, and two coupling-pins, Figs. 11 and 12 (page 612).

The author had referred to "slack gathering" apparatus; in his (the speaker's) opinion the best form of this was the Jones type—the Indian State Railway standard. The Northern Counties

(Mr. Bowman Malcolm.)
Committee's stock had this coupler, which was an exceedingly satisfactory arrangement. It brought the two buffer-heads close together and they were screwed up tight.

The author referred to the question of the circular fire-box. He was sorry that Mr. Aspinall did not touch on that point, because the Lancashire and Yorkshire Railway had had some experience with circular fire-boxes, and it would be interesting to know why their experiments were not carried further in that direction. The difficulty, which seemed to be an insurmountable one, in adopting a fire-box of that type on narrow-gauge railways, was that it was impossible to get a large enough box, unless it was put right above the wheels and of very great length; and he could not conceive how the author would be able to fit engines of the existing type on Irish narrow-gauge lines with boxes of that type.

Mr. G. V. V. Hutchinson said that anyone who had experience in designing modern locomotives for the standard gauge, even the Irish standard of 5 feet 3 inches, would understand the difficulties experienced in designing locomotives for narrow-gauge railways The fire-box was the chief trouble. The main object aimed at should be to keep the fire-box behind the coupled wheels. By so doing, with the 3-foot gauge, and the frames outside the wheels, it should be possible to get a fire-box about 4 feet wide. By placing it between the wheels not only was the grate area restricted, but the water space was cramped. The author had pointed out that the fire-box plates suffered from the scouring action of the fire which had to be forced owing to the small grate area. He (Mr. Hutchinson) thought that the destruction of the plates was also considerably increased by the narrow water-space which prevented good circulation round the box. That would appear to be the case from the experiments that had been made with the circular fire-box. The author stated that he had got rid of most of his fire-box troubles by using the circular fire-box. Although he had a comparatively small grate area, he must necessarily have a good water-space and plenty of room for circulation about the fire-box. Although the author was at present successful with the circular fire-box, he thought that the

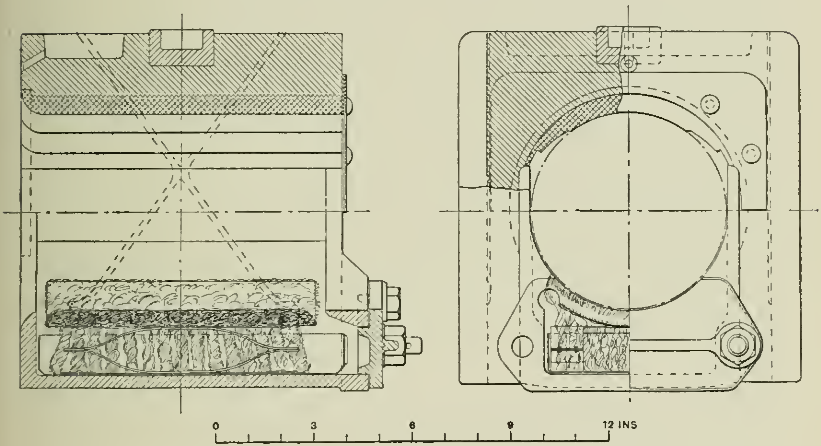
grate area of such a box would soon reach its limit in size, unless a large and heavy boiler were used which might not be desirable for tank-locomotives on a narrow-gauge railway.

The author mentioned that one of his boilers—he believed on Class 4 engine—was not good at steaming. Personally, he wondered whether that was due to the height of the blast-pipe, because from the figures given in Table 1 it should be as good as the others.

Fig. 57.—*Bogie Axle-box and Keep, showing Lubrication from below.*

10 in. × 6 in. Journal.

Great Southern and Western Railway.




The cylinders were an inch larger in diameter, but he did not think that should affect the boiler to such an extent. He noticed in the diagram of the engine (page 608) and in the photograph given that the chimney appeared to be of larger diameter than the other chimneys shown. If the blast-pipe were at the same height as those in the engines with the small diameter chimneys, the blast might not properly fill the large diameter chimney, and it would therefore not create a proper draught on the fire. He would like to ask the author if he had tried lowering the blast-pipe in that engine so as to get the chimney better filled with steam.

(Mr. G. V. V. Hutchinson.)

The author had introduced a considerable refinement in his lubrication by applying forced lubrication to all axle-journals and bogie-centres. Provision was made on the bogies that he (Mr. Hutchinson) was familiar with, on the standard gauge, for lubrication, but the enginemen never troubled to lubricate the bogie-centre at all. When an engine left the shops all the moving parts of the bogie were well coated with black tallow, and that was all the lubrication the bogie-centre received until it came to the shops again. The engines did not seem to suffer much by that treatment. With regard to the lubrication of the bogie axle-boxes, the latest practice on the Great Southern and Western Railway was to do away with the oiling from the top and to do all the oiling at the bottom, Fig. 57, and, since that practice had been adopted, hot bogie axle-boxes were almost unknown. It was not merely necessary to oil from the bottom, but the white metal bearing must be made narrow. Even though the bearing pressure was increased, the box kept cool because the oil had a much better chance of getting up round under the bearing. The narrow bearing was first tried on some of the carriage axle-boxes of the Great Southern and Western Railway, which previously had wide bearings and were continually running hot. But when the narrow brasses were substituted they gave complete satisfaction by being quite cool in running.

Mr. JAMES D. TWINBERROW said the prominent characteristic of the Irish narrow-gauge railway system appeared to be an absence of all system from the standardization of the leading dimensions. The width of gauge indeed seemed to be the only dimension which was standard for almost all.

As the differential length of the inner and outer rails on a curve varied with the distance between them, it was generally understood that the adoption of smaller gauge permitted a proportionate reduction to be made in the length of radius of the minimum curve, and that he presumed was the leading reason for selecting a narrow gauge for a light railway. The author stated (page 601) that the super-elevation was relatively greater for the

narrow gauge than for the broad gauge for the same speed, and presumably for the same radius of curve. As those two considerations determined a definite angle of elevation, one would have expected the actual lift of the outer rail above the inner to be strictly proportional to the gauge. In that respect Irish practice appeared to be against the law.

The effective employment of tank-engines became impossible when, as in the case of many of the Indian 2-foot 6-inch gauge lines, the provision of the necessary boiler power to take care of increasing train loads with longer hauls compelled the fuel and water supply to be carried upon a separate vehicle. He believed that, for many of the Indian lines of that gauge, tank-engines were no longer ordered. The tender-engines usually had leading and trailing trucks with all the coupled wheels in front of the wide fire-box. For mechanical reasons, he thought it was preferable that the bearings of the driving wheels should be inside, whereas the truck wheels might have their bearings outside and thus secure the benefit of the greater width of spring base and better accessibility. When the Hall type of crank was used for the coupled axles, the abnormal dimensions and the difficulties of lubrication, which were referred to by the author (page 603), were experienced.

With regard to the detail of fire-box construction referred to (page 602), he thought it would usually be found that the design involved serious secondary stresses, causing bending on certain parts of the tube-plate and upon some of the water-space stays. It appeared rather remarkable that apparently no effort was made to eliminate the secondary stresses in the drawing office without cost, whereas their existence was a very important factor—indeed, the most important factor in causing continuous expenditure in the repair shop. The width and depth of ash-pans were frequently very restricted, and it became a matter of difficulty to force through a reasonable quantity of air to maintain efficient combustion. An engine which was burning only half a ton of coal per hour required 6 tons of air in the same time, or putting it in another way, about 2,600 cubic feet of air per minute, and in order

(Mr. James D. Twinberrow.)
to get that through the very small ash-pan openings that were frequently provided meant a considerable fall in the air pressure, necessitating a very strong draught. In view of the necessity for maintaining the maximum efficiency of draught, it was noteworthy that British makers apparently did not fit the helicoidal blast-pipe which was now the standard in so many European countries; nor did they appear to have tried the very rational forms of spark-arrester and smoke-box diaphragm plate which were largely used with marked success in Continental practice.

The author appeared to commend the use of a two-wheel bogie or radial truck (page 602). The opinion was generally held—and he believed it might be supported by theoretical considerations—that a single-axle truck, unless it was combined with the next coupled axle in the form of a Zara or Helmholtz bogie, was not a desirable fitting, seeing that it was liable to derailment at K-crossings and elsewhere, and that it was also a cause of severe lateral pressures tending to spread the road.

The earning capacity of a locomotive was proportional to its boiler power, and the cost varied approximately with the weight. A useful figure of comparison was obtained by dividing the weight in tons of the engine when empty into the area of heating surface in square feet. The quotient should be 25 to 27 for tank-engines and from 33 to 35 for tender-engines per ton of engine only. The engine shown in Fig. 5 (page 608), which was referred to as a bad steamer, had only 18·9 square feet of surface per ton, and the duty as specified in Table 1 (page 624) involved the production of about 450 i.h.p. Evidently the boiler could not cope economically with this demand. Fig. 17 (page 613) had 26·4 square feet per ton, but the area of the heating surface was 78·5 times that of the grate surface, and combustion would need to be forced in order to get full output. Fig. 19 (page 614) had 24 square feet per ton of weight and 59 square feet per foot of grate, and should be capable of giving the specified duty without undue forcing.

The tender-engine, Fig. 18, had the very limited axle-load of 6 tons 12 cwt. A considerable saving in weight, cost and frictional resistance would have been effected had the 8½-ton

axle-load been permissible. The engines shown in Figs. 25, 30 and 33 all exceeded 24 feet per ton weight, but the two latter examples could not get round a moderately sharp curve, probably on account of the type of radial box in use.

The outside valve-gear as fitted by British makers was not readily accessible for oiling and overhaul, the centre lines of the valve, its spindle, the expansion link and its connecting-rods rarely fell in one vertical plane; there was consequently an appreciable amount of cross bending on all parts, necessitating additional weight, more lubrication, and shorter life. Despite the advantages consequent upon the use of materials and workmanship of the highest grade, the detail of British designs too often necessitated excessive cost of production, high dead-weight, inaccessibility of parts, and limitation of useful output.

Mr. Livesey, in reply, thanked the members for the very friendly way in which they had received his Paper. Mr. Darbishire had referred (page 631) to his early association with the engines of the County Donegal Railways, and it was a pleasure to him (the author) to meet the "godfather" of his Company's first engines and to be able to say how well they had done their duty. With regard to the cost of narrow-gauge lines, he would deal more fully with that subject in his reply to Mr. Aspinall. Mr. Darbishire did not seem to be quite correct when he said that a good many of the broad-gauge lines were altered to narrow-gauge, there being only three, the original Londonderry and Lough Swilly, 12 miles, the Finn Valley (now part of the County Donegal Railways), 13¾ miles, and the Cork, Blackrock and Passage Railway, 16 miles, and these alterations were consequent on the proximity of other existing and proposed narrow-gauge lines, with which it was considered more convenient to have a physical connection.

Mr. Aspinall (page 633) stated that he did not think any narrow-gauge line in Ireland had been built for less than £6,000 per mile (presumably exclusive of rolling-stock); possibly the average of all might have exceeded this sum, but the author now gave a few figures to show that this extravagant amount had not

(Mr. Livesey.)
always been exceeded. The various sections which make up the system with which the author was connected averaged £5,000, £4,650, £5,894, £4,641, £7,690, £9,337 and £11,500 per mile respectively; only three of these exceeded £6,000, and the first referred to a broad-gauge line, which was altered to narrow in 1894.

The author had remarked that no line should be built as a narrow-gauge if the cost would exceed £5,000 per mile, and he was very strongly of opinion that a substantial narrow-gauge line, with 60-lb. rails, could be built for a sum not exceeding £4,000 per mile (exclusive of rolling-stock), and in spite of the burden of the Board of Trade requirements, provided, of course, that there were no great difficulties due to the nature of the country through which the line was to pass. Such difficulties were rare in Ireland. There was no doubt at all that the Irish narrow-gauge lines had cost far too much; to the author's own knowledge thousands of pounds had been wasted on totally unnecessary and far too elaborate works, and many thousands had been squandered in payment of absurd prices for land, which, in many cases, had been practically valueless until the railway had come along. If the nature of the country and other circumstances rendered a narrow-gauge line absolutely necessary, then there was little prospect of even a moderate dividend being earned if the cost exceeded £4,000 per mile. Parliamentary and other incidental and preliminary expenses were out of all proportion to the value of the work done, especially when there was opposition. This procedure could and should be greatly simplified and the cost reduced to a fraction of what it usually was. The author thought that the consideration of all proposed railway schemes should be dealt with locally. An experienced man, preferably a railway officer, should be appointed, say by the Railway Commission, and after taking evidence from both sides, this officer should report upon it, with his recommendations; and then the Railway Commission, or other authority, could promulgate the decision, which should have the force of an Act of Parliament or of an Order in Council. A procedure somewhat similar to the above was followed when a local authority was desirous of proceeding with some scheme of public utility, when the Local Government Board appointed an officer to hold an inquiry locally, etc.

Good work could be, and was, done by narrow-gauge lines, but what the author maintained was that equally good work could be done on a broad-gauge line and quite as economically, both from the constructional and working points of view. For instance, as far as the gauge was concerned, the cost of a locomotive depended largely on its weight; equally as light an engine could be put upon the broad-gauge, that is to say, no more material need be used in one case than the other. There then need be no appreciable difference in cost, and they had the very material advantage of greater simplicity, accessibility, and reliability, as well as the incalculable advantage of uniformity of gauge. As broad-gauge vehicles could be made to negotiate curves as freely as the others, where then was the advantage, especially in Ireland? When a narrow-gauge tank-engine exceeded 50 tons in weight, as was now necessary on at least two Irish lines, the author failed to see any advantage in the narrow-gauge, rather, much the reverse; the weight of engine above mentioned equalled that of many broad-gauge tank-engines, without any of the compensating advantages, and the cost would be the same in either case. The author could prove his case conclusively, but much of what he would wish to say was so obvious that it would only be waste of space and time.

The system with which the author was associated was broad-gauge in everything except in the actual distance between the rails; broad-gauge loads had to be dealt with and an attempt made at broad-gauge speeds, in order to meet the public demands. The speed limit was really a very serious drawback in the effort to develop the country and compete with existing broad-gauge lines. The author, of course, recognized that a narrow-gauge line was advantageous and, perhaps, really necessary in a few isolated cases, but they were exceptions, to be dealt with as such when they arose; and he agreed with Mr. Aspinall that any future extensions of Irish railways should be made on the standard gauge, but on the lines of a really "light" railway.

Mr. Malcolm was correct in saying (page 635) that a serious mistake had been made, when no standard was laid down for even the height of buffer centres. The author regretted that Mr.

(Mr. Livesey.)
Aspinall did not enlighten the members on his reasons for not extending the use of the circular fire-box, and concluded that it was on account of the difficulty in getting one sufficiently large into the comparatively small space available. It certainly would not be easy to fit such a fire-box to existing narrow-gauge engines, especially in the larger ones, but it could be done if the engine was designed for the purpose.

In reply to Mr. Hutchinson (page 636), the author would go further, and say, that if the fire-box was kept behind the coupled wheels there was no reason why it could not be even six or seven feet wide, according to the limit of running-gauge. The author, from his own observations, had not found that the width of the water space had a very material influence upon the life of plates, so long as this space was a fairly rapidly increasing one from the bottom upwards. In the circular fire-boxes used by the author, when abroad, the grate area was actually rather larger than in the original fire-box, while the heating surface above the grate was reduced; that surface below was more or less useful as such and, no doubt, to some extent compensated for the reduction above. In his (the author's) experience the boilers as fitted with the circular fire-box were more efficient than in the original form; steam was actually more easily made and there was a decided decrease in coal consumption.

It might be of interest if the author gave a few more details of his experiences, which led him to adopt the circular fire-box. The water which had to be used was exceptionally bad, as the analysis (page 645) would show.

It was found that copper fire-boxes of the usual form bulged and fractured so badly at the tube-plate, sides, and back-plate, that they were unsafe to use after about six months' work. Low Moor iron was then tried with steel tubes, with the result that the working life was lengthened to about 12 months; steel of the mildest quality, $\frac{3}{8}$-inch plate, was then used with a resulting life of about 18 months. Although the steel fire-box of ordinary form gave a much longer life under the conditions of working, it was not considered satisfactory enough, and the author ultimately decided to try the

| — | No. 1 Supply. | No. 2 Supply. |
|---|---|---|
| | Grains per gallon. | Grains per gallon. |
| Silica . . . . . . . | 0·896 | 2·420 |
| Calcium carbonate . . . | 4·236 | 6·241 |
| Calcium chloride . . . | — | — |
| Calcium sulphate . . . | 28·056 | 31·718 |
| Magnesium sulphate . . . | 6·972 | 0·533 |
| Magnesium chloride . . . | 52·730 | 56·250 |
| Sodium chloride . . . . | 183·642 | 147·423 |
| Sodium nitrate . . . . | 2·235 | — |
| Total grains per gallon . | 278·767 | 244·585 |

circular form. The first was so good that it was decided to fit it to all boilers, and this was done as rapidly as possible. After several years' work this form of fire-box was found to be as good as new. It was necessary to draw the tubes occasionally in order to clean the interior, as a large amount of scale accumulated, which was not easily removable in the usual process of washing out. One of these engines could be brought into the shop at 3 p.m. on Saturday, after the week's work was done, the fire-box, tubes, etc., removed, and the whole put back after the interior had been thoroughly cleaned, and the engine ready for work at 6 a.m. on Monday. After these boilers were brought into use the author was able to keep 100 per cent. of the engines in regular running for a long period, and this fact saved considerable capital expenditure on the purchase of additional engines, which would otherwise have been required. As regards Class 4 engines of the County Donegal Railways, Mr. Hutchinson wondered whether the poor steaming was due to the height of blast-pipe not being right. In reply, the author wished to state that experiments had been made with many different heights and diameters, without effecting any improvement.

With regard to Mr. Hutchinson's remarks on the lubrication of

(Mr. Livesey.)
the engines illustrated by Fig. 7 (page 608), the author was induced to provide the "refinements" mentioned in order to save time in running and in the preparation of engines for the road. On the earlier engines of this Committee the provision for lubrication was insufficient, and it necessitated constant watchfulness on the part of enginemen. It was the custom for the men to run round the engine at every stop with an oil-can, and frequently this practice was blamed for delays, though not generally the case. The author therefore devised a scheme whereby the time and attention of

*Pressure Lubricators for Bearings, Figs.* 58, 59 *and* 60.

FIG. 58.—*Needle-Valve controlling water supply from boiler.*

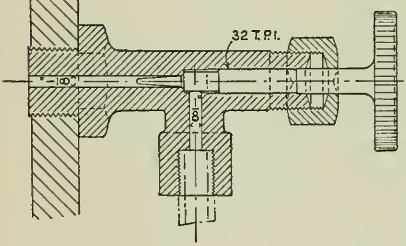
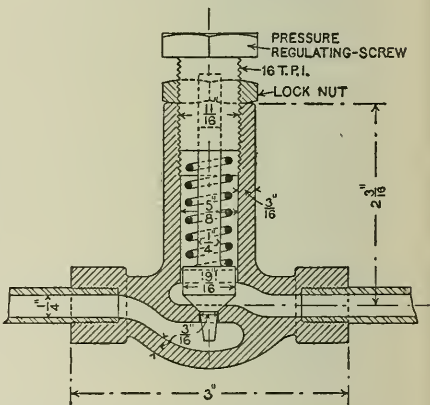



FIG. 59.—*Reducing Valve.*




enginemen was reduced to a negligible quantity. The lubricator consisted of a cylinder, placed in any convenient position in the cab, and which contained the lubricant (in present instance 12 to 14 days' supply); it was fitted with an inlet at the bottom for water, under pressure from the boiler, taken from below water-level, and an outlet at the top for the lubricant to pass. A gauge was also fitted to the cylinder-top to indicate the pressure on the lubricant, and a filling-plug was provided on the top cover. The water from the boiler was controlled to a nicety by means of a small needle-valve, Fig. 58, and passed through a simple and small reducing valve, Fig. 59, by which means the pressure on the lubricant could be adjusted

from full boiler pressure down to, say, 3 to 5 lb. pressure per square inch. It was found that the latter pressure was ample for all purposes. From the cylinder the oil was led by $\frac{1}{4}$-inch galvanized wrought-iron piping along each side of the framing, above the axle-boxes, a tee being fitted opposite each axle-box and a flexible branch pipe, $\frac{1}{8}$-inch bore, led to each box, Fig. 60. At the tee was placed a small needle-valve to control and regulate the supply to each bearing; this only had to be adjusted once for all. Sight glasses were

*Pressure Lubricators for Bearings, Figs.* 58, 59 *and* 60.

FIG. 60.—*Needle-Valve controlling oil supply to axle-box.*

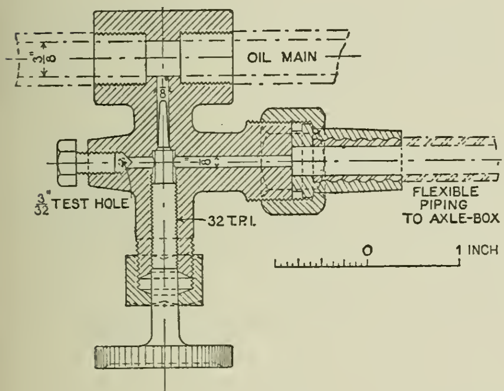
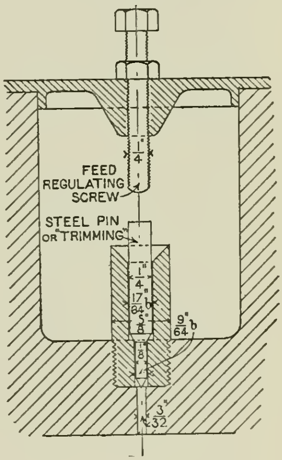



FIG. 61.—*Side-rod Oil Cup and Lubricator.*




fitted near the top of the cylinder for the purpose of showing when the oil required replenishing, and as an additional "refinement" a supply of oil was led to bogie centres and slides, because it was found that it was required, but it was not turned on continuously.

An advantage of the above-mentioned lubricator was that a thick or semi-solid lubricant might be used, and it was independent of temperature conditions. The side-rods were also fitted with a simple device of the author; instead of the usual worsted siphon, or trimming, a solid steel "trimming" or rod was substituted, Fig. 61. The feed was regulated by the rise and fall of this rod,

(Mr. Livesey.)
and was controlled by an adjusting screw fitted in the cover. The feed could be adjusted very minutely, and of course when the engine was at rest the oil-supply was automatically cut off. The complete system mentioned above had been working on three engines for about six months, and had effected a considerable economy in oil, the consumption being the lowest on record for these lines. There was an entire absence of splashing about the side-rods and wheels, so noticeable in other engines, and, what was more important, a great saving of enginemen's time; the side-rods did not need attention more than twice weekly, and the axle-boxes once a fortnight. The needle-valves were illustrated in Figs. 58 and 60; the cylinder needed no description, as it could take any form, suited to the circumstances under which it might have to work. The water-supply pipe was fitted with a two-way cock at the foot of the cylinder, one way being to waste, in order to empty the cylinder of water when required to refill with oil.

The author agreed with Mr. Hutchinson that oiling from the bottom was preferable, where possible, all his newer engines having bogie axle journals so lubricated, and practically all his carriage and wagon stock had now been fitted to lubricate from the bottom. A hot journal, with this type of axle-box, was unknown on the County Donegal lines. The author's practice was to make the width of bearing one-sixth of circumference.

In reply to Mr. Twinberrow (page 638), the author did not agree with his suggested reason for the selection of a narrow-gauge for a light railway. It was possible by suitable design, and, in some cases, by widening the gauge slightly, according to rigid wheel-base, to get broad-gauge vehicles to run freely round any curve. Mr. Twinberrow appeared to be wrong in his expectation that the actual lift of the outer rail on curves should be strictly proportional to the gauge for the same speed and curve. Was not "inversely" proportional the more accurate word to use? No doubt tender-engines were necessary in some circumstances, but they should never be used if it was possible to design a tank-engine to do the work.

The author differed entirely from Mr. Twinberrow in regard to the position of the bearings, which should always be outside the wheel, whatever form of crank was used, a very much steadier engine being the result. The author had had no difficulties with the lubrication of Hall crank-bearings. He agreed with Mr. Twinberrow in regard to fire-box design, but from many years' personal experience of the radial truck, and from the fact that it was largely and successfully used in foreign practice, he was convinced that it was equally as safe and satisfactory as the four-wheeled bogie; he had never known of a derailment due to the use of such form of truck. He (the author) failed to see how the two-wheeled radial truck could exert greater lateral pressure upon the rail than the four-wheeled bogie, though, of course, the pressure was confined to one point instead of being divided over two. The total pressure, tending to spread the road, would be the same for both forms, the conditions under which they worked being equal.

The author hardly agreed with Mr. Twinberrow's final remark that the outside valve-gear, as fitted by British makers, was not readily accessible for oiling and overhaul. The fact that it was outside was a great point in its favour, for the purposes named, and there appeared little difficulty in eliminating all tendency to cross bending, though no doubt weight and cost might be lessened with advantage in many cases.

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*Communications.*

Mr. Herbert W. Garratt wrote that he was quite in agreement with the author as to the difficulties imposed upon locomotive design and capacity by the narrowness of gauge, and also by a limited load-gauge. As stated in the Paper, it was a fact that the fire-box was one of the greatest difficulties, and the objection he mentioned as to firing and the damage caused to the side sheets due to the narrowness of the fire-box were actual and serious facts,

(Mr. Herbert W. Garratt.)
and undeniable, but the objections no longer existed if the locomotive were of the type below referred to. The fire-box in the "Garratt" locomotive was not restricted as to width by any other consideration than that of the limitations of the load-gauge minus space for the frames, nor for length either, and it might be constructed to dimensions, say twice the width of the gauge, and as long as required without limit (*see* the tank locomotive for the Tasmanian Government Railways, Fig. 62, Plate 26). The weak point, referred to by Mr. Livesey, was therefore eliminated, and the fire-box in the writer's locomotive could be as commodious as the 3-foot gauge carriages to which he referred. Many engines of this type were running on various railways in different parts of the world, giving entire satisfaction; some up to 94 tons weight for the 3-foot 6-inch gauge had just been built, and others were under construction both here and on the Continent.

In 1911 six engines, weighing 66½ tons each, with a tractive effort of 25,200 lb. (limited to a maximum axle-load of 9 tons), were supplied to the Western Australian Government Railways, 3-foot 6-inch gauge. These had proved so satisfactory in every way that a further order for seven more of the same type had been placed by the same Government. There need be no more trouble with narrow-gauge engines, either as regards size and shape of fire-box, height of centre of gravity, distribution of weight, or stability on curves; and, furthermore, in the "Garratt" type the total weight of the engine was distributed over a greater length of rail than in any other. Other advantages included freedom of running, thus reducing cost of permanent-way maintenance and tyre renewals, etc.

Mr. LIVESEY wrote that he was glad to hear of the success of the "Garratt" locomotive, as he thought that type eminently suitable for heavy switchback work, particularly on narrow-gauge lines and where sharp curves abounded. In many instances the limit had been reached for locomotives of the ordinary type, but Mr. Garratt had opened up a wide field of increased usefulness by his ingenious design.

The author mentioned in the Paper that he might be able later to give some particulars of the performance of the new superheater tank-engines of the County Donegal Railways; and he was now in a position to do so. There were three of these engines, which had been put to work on the heaviest sections of the line. One of them worked between Killybegs and Stranorlar, and the other two between Killybegs and Glenties, via Stranorlar. A glance at the profile of line, Fig. 1 (page 605), would give an idea of the heavy work required of them. They had been given some very heavy work to do, and recently an excursion party of 1,500 was conveyed, in two trains of fifteen coaches, from Londonderry to Ballyshannon; one of the new engines hauled each train, which was estimated at 230 tons exclusive of engine ($50\frac{1}{2}$ tons). The run from Derry to Stranorlar was performed in 60 minutes, no effort being made to run fast, because the train was ahead of time all the way. From Derry to Donegal the trip was a non-stop one, the portion from Stranorlar to Donegal being run in 53 minutes, the speed up the bank being steady at 12 miles per hour, and a superheat of 700° being easily maintained. Equally as good a run was made on the return trip, though the speeds were higher, as there was a clear road all the way, the average speed between Stranorlar and Derry being 35 miles per hour. The smoothness of running of these engines was most remarkable.

The superheater had added fully 30 per cent. to the capacity of the engines; the coal-consumption had fallen fully 20 per cent., and the water-consumption about 30 per cent. The actual coal burnt per mile (Cumberland coal) worked out at 29 lb., and the author considered that with good Welsh coal this figure would be between 24 and 25 lb. The average of the earlier engines, Class 5, Fig. 6 (page 608), with the same coal, was between 35 and 36 lb., and for Class 4, Fig. 5, it was between 44 and 45 lb., although these latter engines had much lighter work to do. The improvement in the efficiency of the machine was made very evident from the following: The new engine was able to haul a gross load (including engine) of more than double that of Class 1, the actual ratio being 11 to 5 on the same coal- and oil-consumption per mile,

(Mr. Livesey.)
though at higher speeds, or, in other words, the cost of hauling a ton had been reduced to one-half. As far as the superheater was concerned no difficulty whatever had been experienced; the men had mastered its use easily and quickly, and in practice it had been found quite easy to maintain a superheat of 700° F. With light loads the temperature was maintained at about 650° F. without trouble; it dropped to about 450° F. during a stop, but quickly rose again after starting.

An interesting fact had been learnt in regard to cylinder lubrication: it was that no more oil was needed for the superheater engine than with the non-superheater types, the average for the former being the same, namely, 0·9 pint per 100 miles. The author thought that the frequent stops and the numerous occasions on which the engine ran without steam, while the lubricator pump was still forcing oil to the various parts, accounted for this. The advent of these engines had effected a decided improvement in the timekeeping of trains, though at no time was it ever seriously at fault.

# NEW GRAVING DOCK, BELFAST:

## Mechanical Plant and General Appliances.

By W. REDFERN KELLY,

Engineer-in-Chief to the Belfast Harbour Commissioners.

The new Graving Dock at Belfast, Fig. 1 (page 654), is one of the finest and best equipped dry docks in the world at the present moment; and is indeed the only graving dock in which it is possible to place the leviathan steamer, the "Olympic," the world's largest existing specimen of naval architecture. It is not only famous for its huge capacity, but for its general equipment, which is of the most modern and up-to-date type. The Belfast Harbour Commissioners, who have at all times manifested a keen desire to facilitate, in every legitimate and reasonable way, the great shipbuilding industry (which may be considered to be *the* staple industry of this port and city), have, in order to further that particular trade, and to encourage the Admiralty to place with our local shipbuilders a fair proportion of their orders, expended upon this great dock and its collateral works, a sum of £350,000.

The works were commenced early in the year 1904, and the time required for their construction was about seven years. This period might, however, have been greatly reduced, were it not for the many engineering difficulties which were encountered, by the

Continued below.

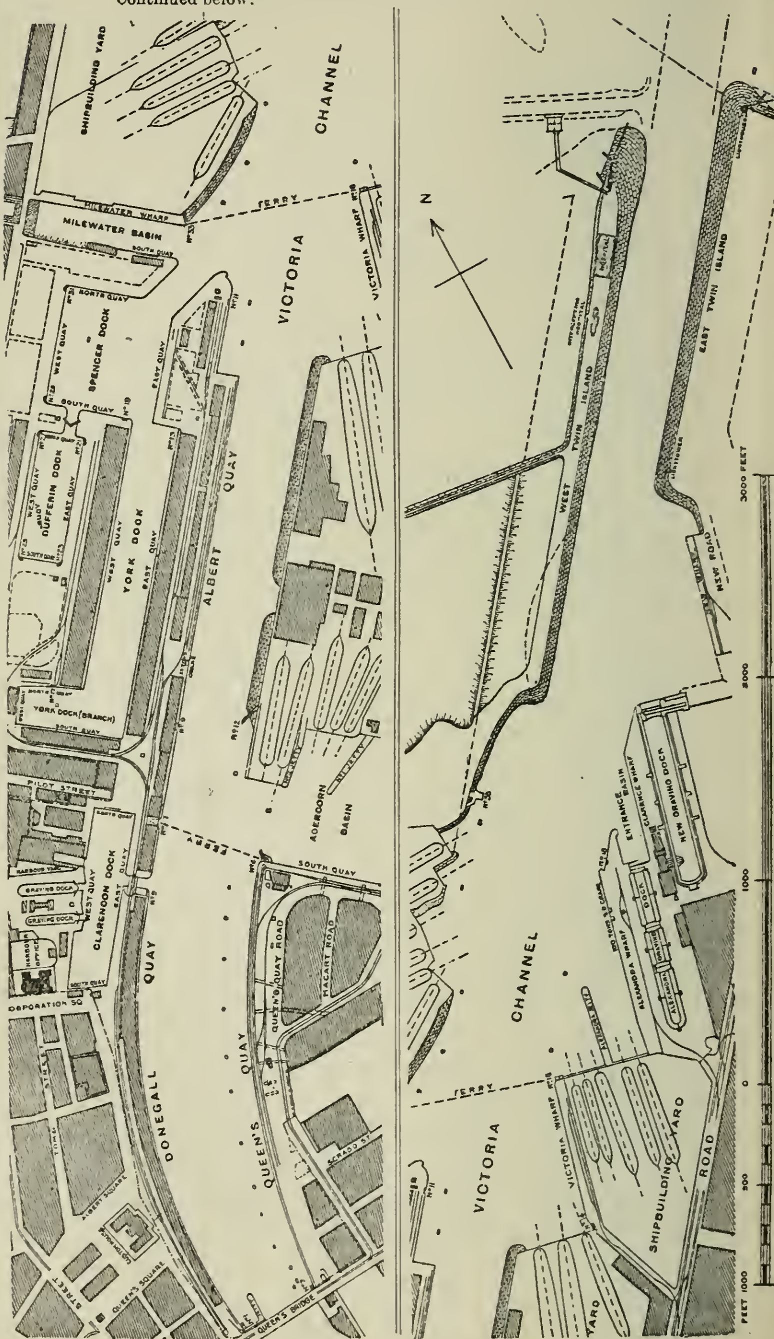



FIG. 1.—*Plan of part of the Port and Harbour of Belfast.*

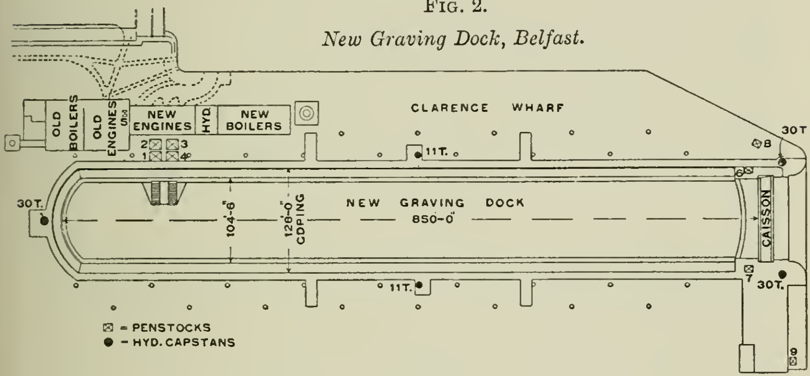
Continued above.

contractors, during the progress of their operations. Although no formal or ceremonial opening of the new dock has as yet taken place, it was nevertheless commissioned for practical use on the 1st April 1911, when the S.S. "Olympic" was admitted for final fitting out purposes, a few weeks immediately prior to her departure on her maiden voyage.

*Dimensions of the Graving Dock.*—The general dimensions of the dock, Fig. 2, are as follows:—

Fig. 2.

*New Graving Dock, Belfast.*




Length of dock, on floor, from the inner face of the caisson to the toe of the battered wall at the south end of the dock, 850 feet; or if the caisson be placed in its outer berth, about 887 feet. Length, over all, that is, from the coping at the south end of the dock to the inner face of the caisson when in its outer position above the apron, at the entrance, 901 feet.

Breadth of dock, from toe to toe of the battered side-walls below the altar courses, 100 feet; from coping to coping, 128 feet; and at entrance, 96 feet.

Depth of surface of floor of dock, at its centre, below harbour datum (which latter is 1 foot 9 inches below average low-water line), 26 feet 6 inches. Height of coping, above harbour datum, 16 feet. Depth of floor surface (at its centre) below the level of

the entrance sill, 2 feet. Height of keel blocks above the floor 4 feet 6 inches; or without timber capping, 3 feet 6 inches.

| | |
|---|---|
| Depth of water on keel blocks, at H.W.O.S.T. . . | 32 feet 9 inches. |
| ,, ,, ,, ,, dock sill, ,, ,, . . | 35 feet 3 inches. |
| Width of caisson chamber, in the clear . . . | 23 feet $4\frac{1}{2}$ inches. |

The dock floor falls 12 inches on either side of the centre line, towards the open drains at the sides of the dock, no longitudinal fall being given.

The level of the surface of the inner and outer sills, at the centre, is 40 feet 6 inches below coping line, or 24 feet 6 inches below harbour datum. The level of the surface of the concrete apron, northward of the outer sill, is 43 feet 6 inches below the coping level; or 27 feet 6 inches below harbour datum. The level of the upper surface of the brickwork invert of the caisson chamber, and the caisson track, in the centre, is 45 feet 3 inches below coping level, or 29 feet 3 inches below harbour datum.

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The Paper is not intended to deal with the general construction of the Graving Dock proper, but rather with those items of mechanical plant, such as the pumping installation, hydraulic system, boilers, capstans, caisson, and caisson hauling plant (together with its hauling appliances), culvert sluices, etc.; as such details of dry-dock equipment naturally possess much greater interest for the mechanical engineer than would the structure of the dock itself, with which latter the civil engineer is necessarily more intimately concerned; and the author hopes that the importance of the subject may be deemed sufficient to justify the demands which he has been obliged to make upon the space usually allocated to such contributions.

## Pumping Appliances.

The capacity of the Graving Dock, below the level of high water of average spring tides, without making any allowance for displacement by a vessel placed in the dock, is about 21 million gallons of water; and the duty imposed upon the pumping plant

is that of discharging this great quantity of water, within a period of not more than 100 minutes from the time of commencing the pumping operations. During this pumping period the tide outside the dock will usually fall to the extent of about 2 feet.

*Pumps.*—The plant provided for the performance of the above work (*see* Plan, Fig. 3, Plate 27; and Elevations, Plate 28) comprises three main pumps of the centrifugal type, which are driven by three cross compound vertical engines. Each pump has two suction-pipes, 42 inches in diameter, and one delivery-pipe, which tapers from 54 inches to 60 inches in diameter. The impeller, or pump-disk, is 7 feet 6 inches in diameter. The extreme depth of the pump-casing is 12 feet $7\frac{3}{8}$ inches; and the over-all width of the pump is 12 feet 10 inches. The centre of the impeller-disk is 18 feet $3\frac{1}{2}$ inches, and the top of the pump-casing is 24 feet $\frac{1}{2}$ inch respectively, above the dock floor, close to the inlet to the pump well. The six large suction-pipes are carried through the engine-room floor, into the main sump underneath, their bell-mouthed ends reaching a level of about 6 feet $8\frac{1}{2}$ inches below the dock floor, close to the inlet, and 5 feet $3\frac{1}{2}$ inches above the floor of the main sump. The pipes are securely cement-grouted against the foundation brickwork through which they pass, and have thus been made perfectly water-tight.

The pump-casings, as well as the outer covers, are so constructed as to be easily removed for overhaul without disturbing the suction or discharge-pipes. Suitable sight-holes, with covers which may be easily removed and replaced, are provided on each side of the pump-casing on the top of the suction; and provision is made for getting at the interior of the pumps to admit of their being examined, and to give easy access to the keys securing the disks to the pump-shaft. The pump bearings are of white metal, one on either side of the disk, and are each 3 feet in length.

Each pump is fitted with a steam-ejector, in order that the air can be exhausted in the pump-casings and suction-pipes, when restarting the pumps after their having been stopped, and when the water has been lowered a depth of about 4 feet in the dock.

The discharge from the ejectors is led into the main pump discharge-culvert. Each pump is also provided with a water-gauge, to indicate when the pump-casing is full. To each of the main pumps a sluice-valve, 54 inches in diameter, worked by hydraulic power, is fixed to the delivery branch, to which the main delivery-pipe is bolted. These discharge-pipes, three in number, are carried through the side wall of the engine and pump-room into the outlet culvert, through which the delivery is led to the tideway at the south end of Clarence Wharf. At the outer end of each delivery-pipe a properly balanced flap-valve, of elm timber, known as a "foal's foot" valve, is fixed, and can be approached through a covered man-hole, just above the valve, outside the engine-house.

For the purpose of dealing with any leakage, or drain waters from the dock culverts, which the main pumps cannot reach, an auxiliary pump has been provided. This pump has suction and discharge-pipes, each 14 inches in diameter. The diameter of the impeller, or pump-disk, is 48 inches. The pump is driven by an inverted direct-acting compound cylinder engine.

*Engines.*—There are, in all, for the main pumping (*see* Plan, Fig. 3, Plate 27; and Elevations, Figs. 4 and 5, Plate 28), three cross compound non-condensing vertical engines, one set to each main pump, coupled direct. The cylinders of these engines are 22 inches and 38 inches diameter, respectively, with a stroke of 20 inches; they are capable of running at an average speed of 125 revolutions per minute, and at this speed the engines run quietly, due to proper steam distribution and efficient balancing. They are designed to suit a working pressure of 160 lb. per square inch (100° F. superheat); and each set develops 741·5 i.h.p. at 121 revolutions per minute. They are steam-jacketed around the top, bottom, and sides, and are covered with an asbestos non-conducting composition, and cased with steel planished sheets. A water service has been provided for all main bearings.

The cylinders, with their jackets, are all automatically drained, by means of pipes and steam-traps, so that danger of water

hammering is removed, and no handling of the drains by the attendant is necessary. Main stop-valves are balanced, and can be worked with great ease, which enables engines to be easily handled. The steam-pipe range is of solid-drawn steel tubes throughout, with flanges riveted on, a large steam-separator being placed at the bottom of the vertical pipe which leads from the boiler to the engine-house. Expansion in the pipe range has been so well provided for that there is no difficulty in keeping the joints tight.

The leakage pump is driven by a small direct-acting inverted cylinder cross compound non-condensing engine, coupled direct. This pump runs at about 300 revolutions per minute, and is intended for the removal of such leakage water as may find its way into the dock after the main pumps have ceased to draw.

Among the various accessories to the pumping plant may be mentioned:—

A small duplex pump (Odesse type), with pump-piston $2\frac{3}{4}$ inches in diameter, and 4-inch stroke, is fitted on the engine-room floor, for internal drainage purposes.

An overhead travelling-crane, worked by hand, and capable of lifting weights up to 7 tons, has been provided, and travels just below the roof, Fig. 5, Plate 28.

An automatic water-gauge, with vertical index board, has been erected in the engine-room, in order to indicate with facility the level of the water in the graving dock at any moment. It is constructed on a scale of half full-size. The gauge is operated by a float, acting within a metal pipe, the latter being led to the bottom of the sump; and the float is placed in communication with the indicator by a cord of copper wire, led over reducing pulleys, and to which a pointer is suspended. The arrangement is an ingenious and convenient one, the gauge being visible from all essential points in the engine and pump-room.

The following are some of the leading figures for the pumping plant:—

| | |
|---|---|
| Diameter of Cylinders . . . . | 1 ft. 10 in. and 3 ft. 2 in. |
| Length of Stroke . . . . . | 1 ft. 8 in. |
| Diameter of Piston-Rod . . . | 4 in. |

| | |
|---|---|
| Diameter of Shaft-Journals . . . | $8\frac{1}{2}$ in. |
| Length of Connecting-Rod . . . | 4 ft. 2 in. |
| Diameter of Pump-Impeller . . . | 7 ft. 6 in. |
| Diameter of Pump-Shaft . . . | $8\frac{1}{4}$ in.–$8\frac{3}{4}$ in. |
| Capacity of Feed-Heater, per hour . . | 4,600 lb. |
| Temperature of Feed in degrees F. . | 200°. |
| Evaporation of Boilers, per hour . . | 64,000 lb. |

The pumping plant contract was carried out by Messrs. Andrew Barclay, Sons and Co., of Caledonia Works, Kilmarnock, and cost altogether £12,272 7*s.* 7*d.*

*Hydraulic Power-Supply Plant.*—This plant, Plate 29, comprises two sets of pumping engines and pumps, and a hydraulic accumulator, together with the requisite water pressure and return piping, and is utilized for the work of opening and closing the travelling caisson-gate, the lifting and lowering of the nine penstock sluice-doors, the working of the five hydraulic capstans, and also of the sluice-valves in the engine-room, or to such other purposes as it may be found necessary, or desirable, to apply the available water-power.

The pumping engines are direct-acting, with two inverted cylinders arranged over the crankshaft, and each set of engines operates two bucket and plunger pumps, directly from the piston-rod crossheads. Each pair of engines is capable of developing not less than 166 actual horse-power, when working with steam at 160 lb. pressure per square inch, and running at 58 revolutions per minute ; and the combined capacity of the two pairs of engines and two sets of pumps is that of supplying not less than 630 gallons of water per minute, at 750 lb. pressure per square inch.

The cylinders are two in number, the high-pressure cylinder being 17 inches in diameter, and the low-pressure cylinder 36 inches in diameter, both having a clear piston-stroke of 18 inches. The cylinders and covers are steam-jacketed, the cylinder bodies being covered with asbestos composition, lagged with wood, and cased in with highly planished steel sheets. The pistons are of the Rowan type.

The back columns are of cast-iron, of the box form, and are suitably arranged to receive the guide-bars and pumps, accurately machined and attached to the cylinders and to the bed-plate by bolts and nuts. The front columns, four in number, form the receptacles for the pumps. The bed-plate, which is of the box form, is of cast-iron, ribbed and bracketed throughout. The fly-wheel is of cast-iron, of disk form; it is 7 feet in diameter, and is 12 inches in breadth on the face, being properly balanced to ensure steady running.

The hydraulic pumps are four in number, and are of the bucket-plunger type, each plunger being $5\frac{1}{4}$ inches in diameter; they have a separate gland packed with hydraulic packing. The bucket is $7\frac{1}{2}$ inches in diameter. The bucket-valves, head-valves, and foot-valves, are of the multiple type, each set consisting of nine valves, of $1\frac{1}{4}$-inch diameter. These valves are cut from the solid, and all valves, seats, and guards are cast from a special mixture of hard bronze. The stems at the tops of the valves are prolonged, and work in guides forming part of the guards, each valve being provided with a spiral spring made from brass of high tenacity. The glands for the rams are of polished steel, each being secured by four turned steel studs of $1\frac{3}{4}$-inch diameter, and all gland-nuts are provided with suitable locking arrangements. Each inlet to any of the suction-valves is provided with a stop-valve, and all necessary pipe connections between the pumps and accumulator are provided.

The hydraulic accumulator, Plate 29, is of the suspended type. The ram is 18 inches in diameter, and 20 feet stroke (composed of a special cast-iron mixture for hydraulic work), turned all over, and polished on the working parts. It is provided with a special bayonet arrangement to prevent its being blown out of the cylinders. The body of the ram is $2\frac{1}{2}$ inches thick, well ribbed and bracketed internally, and is cast in one length, vertically. The top end of the ram is fitted with a cast-steel crosshead, which is arranged to receive eight steel suspending bolts, for attachment to the weight casing.

The cylinder, which is of a special mixture of cast-iron, is cast in one piece, vertically, with a heavy sinking head cast on the upper end. The shell is $3\frac{3}{4}$ inches thick. The stuffing-box is accurately bored, and is provided with a bronze neck-ring, and a mild cast-steel gland, with a gun-metal liner, the gland being held in place by eight large turned steel studs and nuts. The bottom end of the cylinder is cast solid, and is turned and faced, to suit the socket on the sole-plate. The sole-plate is of cast-iron, in one piece, well ribbed and bracketed.

The weight casing, which is cylindrical, is 10 feet 6 inches in diameter, and 18 feet in height. It is weighted with a mass of concrete, amounting to about 78 tons. The outer casing is composed of $\frac{1}{4}$-inch steel-plates, lap-jointed, and stiffened by angles and gusset stays, the central tube being made of $\frac{3}{8}$-inch steel-plates. The casing is well stiffened, and specially strengthened at the points for attachment to the sling bolts, and has four cast-iron guide-blocks to work between double angles, which latter are fixed to the vertical timber guide-columns.

Among the accessories may be mentioned a cast-iron suction-tank, 10 feet long, 6 feet wide, and 5 feet 6 inches deep, which is capable of holding about 2,000 gallons of water. The tank is provided with all necessary and usual fittings, including float, gauge, pipe connections, overflow pipes, scour pipes, ball-valves, etc. All parts which are subjected to accumulator pressure have been tested to a pressure of 2,500 lb. per square inch. The pressure-pipes have diameters ranging between 6 inches and 3 inches, and the return piping diameters ranging between 4 inches and 7 inches. These pipes are of cast-iron, composed of a special mixture for hydraulic work. The pressure-pipes are fitted with momentum valves, where necessary. The cost of executing the above works amounted to £4,657 12*s.* 10*d.*

*Boilers, Feed-Heater, and Steam-Piping.*—The steam installation, Plate 30, comprises four Babcock and Wilcox water-tube boilers of the marine type, each having a heating surface of 3,590 square feet and a grate area of 105 square feet. Each boiler

is hand-fired and is composed of thirty-one sections of tubes, each section being eleven rows in height, the tubes being $3\frac{3}{16}$ inches outside diameter by 10 feet 9 inches long. The headers into which the tubes are expanded are made from $\frac{1}{2}$-inch mild-steel plates, and are provided (opposite each tube end) with an oval hand-hole door, having a joint on the inside of the header. The interior examination and cleaning of each tube is thus a very simple matter. The removal of soot from the outside of the tubes is effected by brushes or steam-lances, these being inserted through doors in the sides of the boiler provided for the purpose. There are eight of these doors in each boiler.

The boilers are each provided with a Babcock and Wilcox integral superheater having a heating surface of 550 square feet and being capable of increasing the temperature of the steam by 100° F. at the engines. It is worthy of note that the radiation of heat from the boiler-casings is very efficiently minimized by the fitting of a special non-conducting fire refractory material manufactured by Messrs. Babcock and Wilcox, and named by them "Cellinsulate." This material is insoluble and very light, weighing only 20 to 23 lb. per cubic foot.

The contractors for the pumping plant having stipulated that the boilers should be capable of evaporating altogether at the rate of 64,000 lb. of feed-water per hour, it was arranged that each boiler should be capable of evaporating 16,000 lb. of water per hour. The boilers and superheaters have been constructed for a working pressure of 170 lb. per square inch, and the various pressure parts were tested hydraulically to upwards of 260 lb. per square inch.

The boiler-house being limited in width to 32 feet 6 inches, it became necessary to construct along the interior of this room an overhead flue to carry away the gases of the four boiler furnaces to the chimney. This main flue varies in section between 6 feet by 6 feet and 9 feet 4 inches by 7 feet 4 inches inside the boiler room, and 11 feet $5\frac{1}{2}$ inches by 5 feet $5\frac{1}{2}$ inches outside the building and between the boiler-house and chimney. It is constructed of $\frac{1}{4}$-inch mild-steel plate, lined on the outside with 2 inches of "Cellinsulate"

and protected by $\frac{3}{32}$-inch thick outer plating. The flue plates are connected by angle-bars, and are stiffened by tees, angles, etc., due provision for expansion having been made. The main flue is carried by columns and girders, the ends of which latter on one side rest upon the boiler-house wall. Two duplex direct-acting steam feed-pumps (10 inches by 6 inches by 10 inches) are provided. The pumps draw from the water-storage tank, and discharge into the feed-heater and thence onward to the boilers.

The feed-heater, Fig. 6, Plate 29, which is of the Royle vertical cylindrical type, fitted with Row's indented tubes, is capable of heating the feed-water to about 200° F. at the boiler pressure of 170 lb. per square inch when working at full power. The feed-heater is covered with asbestos non-conducting composition, and is sheeted with planished steel.

The steam-main runs at the back of the boilers, with which they are connected by means of four 5-inch branch pipes. The piping is 10 inches, 7 inches, 5 inches and 2 inches diameter, respectively, and is composed of weldless mild-steel. The bends are of the same material and are constructed one gauge thicker than the straight pipes, so as to allow for the thinning action which would take place during the operation of bending. The flanges are of mild-steel stamped out of the solid and secured to the pipes (in the case of all sizes up to and including 6 inches diameter) by means of a fine screwed thread, the ends of the pipes being afterwards expanded into the flanges; in the case of pipes 7 inches in diameter and over, the flanges are fixed by means of expanding and riveting, the rivet-holes being drilled radially to the centre of the pipe, burrs removed, and rivets driven by hydraulic pressure. In both cases the flanges are provided with machined surfaces, and are drilled to the template of the British Engineering Standard Report, Table No. 2.

The branch connections are formed by means of mild-steel nozzles riveted to the pipes.

The scantlings of the pipes and flanges are as follows :—

| Bore of pipe. | Thickness of pipe. | Thickness of flange, including facing strip. | Radius of bends. | |
|---|---|---|---|---|
| Inches. | Inch. | Inch. | Feet. | Inches. |
| 10 | $\frac{5}{16}$ | $1\frac{1}{4}$ | 3 | 6 |
| 7 | $\frac{1}{4}$ | $1\frac{1}{8}$ | 2 | 0 |
| 5 | $\frac{3}{16}$ | 1 | 1 | 6 |
| 2 | $\frac{3}{16}$ | $\frac{11}{16}$ | | 8 |

An auxiliary steam-main $1\frac{1}{2}$ inches, 2 inches, and $2\frac{1}{2}$ inches in diameter, respectively, is carried over the tops of the boilers, together with the necessary $1\frac{1}{2}$-inch diameter branches from the boilers, and $1\frac{1}{2}$-inch connections to the stop-valves on the pumps. The pipes are of weldless mild-steel, with wrought-steel flanges screwed on and expanded, as previously described for main steam-range. The tee-pieces are of cast-steel, the flanges being provided with machined surfaces and drilled to template. Necessary 2-inch, 3-inch, and 4-inch diameter delivery-pipes, with duplex 2-inch branches, together with the necessary valves, bends, tee-pieces, etc., are provided. The pipes are of weldless mild-steel, with wrought-steel flanges screwed on and expanded. The tee-pieces are of special cast-metal, of cold blast iron and steel, of high tensile strength. The blow-off cocks are connected by $1\frac{1}{2}$-inch branches into a 2-inch main running at the back of the boilers and terminating just outside the boiler-house wall. The pipes are of weldless mild-steel, with wrought-steel flanges screwed on and expanded; the tee-pieces, elbows, etc., are of special cast metal, as previously referred to.

Mild-steel drain-pockets are provided for the steam-main, and connected by weldless steel flanged piping to a 1-inch diameter Geipel and Lange steam-trap. All the steam-piping has been covered with a superior description of compound, non-conducting composition, about 2 inches in thickness, which is covered with cotton canvas.

The contract for the boilers was entrusted to Messrs. Babcock and Wilcox, Ltd., of London, and cost altogether £8,667 3*s*. 8*d*.

*Hydraulic Capstans.*—There are in the dock equipment five capstans which are worked by hydraulic power, three of them being of 30 tons, Plate 31, and two of 11 tons capacity each, Fig. 9, Plate 32, respectively, the water-power by which they are actuated being supplied at a pressure of 750 lb. per square inch. The delivery or pressure-pipes vary between 6 inches and 3 inches in diameter, and the return pipes to the accumulator-house vary between 4 inches and 7 inches in diameter. The three largest of these capstans, Fig. 8, are the most powerful which have ever yet been employed as graving dock accessories at any port in the world, the next largest size in use in dockyards being those adopted by the Admiralty, which are only of 16 tons capacity each.

The capstans are all double-powered, the three largest being each capable of giving a hauling stress of 30 tons direct from the capstan barrel with a single rope at a speed of about 30 feet per minute, while their gearing is so arranged as to give a lower hauling stress of 7½ tons at a speed of about 120 feet per minute. The two smaller-power capstans, Fig. 9, are capable of exerting a hauling stress of 11 tons direct from the capstan barrel, at a speed of 30 feet per minute, and a speed of about 50 feet per minute when exerting the lower hauling stress of 7 tons.

The capstan-heads are fitted with pawls for use when it shall be found necessary to operate the capstans by hand, and for the latter purpose eight hand-spikes of ash have been provided, with proper recesses in the head of each capstan to suit same. All those parts of the capstans which are subject to working pressure were tested to a pressure of 2,500 lb. per square inch before leaving the makers' works.

The capstan contract was entrusted to Messrs. Sir W. G. Armstrong, Whitworth and Co., Ltd., of Newcastle-on-Tyne, and cost £3,454 16*s.* 3*d.*

*Caisson Gate.*—The gate at the entrance to the dock, Plate 33, is of the travelling-caisson type, comprised mainly of steel, and having, at its top, an automatic folding bridge, which, by a parallel-bar arrangement, is lowered as it enters the caisson recess, and raised

when it clears the recess. When lowered, the caisson can be travelled upon rollers into a caisson recess, which latter is roofed over, and is formed into a roadway for vehicular traffic. The caisson is rectangular in shape, one of its sides being longer than the other. The shorter side is intended to bear against the south granite meeting faces of the inner sill of the entrance. The longer side is intended to bear against the north meeting faces of the latter sill, and as well against the granite meeting faces of the outer sill of the entrance. So that when the caisson is in its normal or inner track the length of the dock on the floor is 850 feet, and when it is placed against the meeting faces of the outer sill the length is 886 feet 7 inches. This interchangeability has been found to be of the utmost value in the docking of the S.S. "Olympic" and "Titanic."

The caisson is in length on one side 103 feet 4 inches, and on the other side 98 feet 4 inches. It is in clear width over the greenheart meeting faces 18 feet 4 inches, and in height from the bottom of the keelsons, which bear upon the rollers, to the roadway surface of the folding bridge 42 feet 4 inches, which latter is level with the coping of the side walls of the dock entrance. The caisson is divided into two compartments, the lower of the two being an air-chamber, and the upper a water-ballast chamber; the latter is provided with valves, two on either side of the caisson, by the manipulation of which the tidal water may be admitted or excluded as may be desired and as may be found necessary. When the caisson is in place, in order to counteract its tendency to rise as the tidal rise increases, it may be necessary to open the valves on the seaward side, so that the tide may ebb and flow in the water-ballast chamber, or to close those valves to retain any constant desired quantity of water in the latter chamber for steadying purposes. A water-tight deck divides the above two compartments, and the valves referred to are placed at this deck level, bends being led from them on both sides of the caisson below the level of the dock and communicating with the tideway. These bends are fitted with proper rose heads in order to prevent floating matter from being carried into the chamber. The valves

and their seats are of gun-metal, and the former are controlled from a partial deck about 5 feet below the upper surface of the roadway bridge of the caisson. In order to gain admission to the lower or air-chamber of the caisson, two vertical water-tight tubes or trunks, 30 inches in diameter, are provided, and are fitted with a man-hole and cover, bolted on, and so placed as to clear the folding bridge platform when down. Ladders are fixed, one in each trunk, leading to the bottom of the caisson. Two similar ladders are provided for giving access to the water-tight deck. At the bottom of the air-chamber are stowed the portable ballast-blocks, which are of concrete, 12 inches by 12 inches by 6 inches, and weighing each about 70 lb. Below this portable ballast there is the permanent ballast, consisting of concrete, laid in situ in the ordinary way. The total weight of concrete ballast found necessary amounts to 986 tons, of which the weight of the permanent ballast is 288 tons, and that of the portable ballast 698 tons. With this quantity of ballast the caisson will float when the water-ballast chamber is empty, the draft being 30 feet, and with a depth of water on the entrance sill of 35 feet 3 inches. The structural weight of the caisson is about 455 tons, which, together with all ballast, amounts to a total weight of 1,441 tons. The caisson, when in its normal track, rests, by its two longitudinal steel keelsons of 8 inches by 4 inches section, upon fifty-two cast-iron rollers placed in cast-iron roller-boxes, which latter are built into the floor of the caisson track. The roller spindles are of mild steel lined with gun-metal.

When the lengthening of the dry-dock space, beyond 850 feet on its floor, becomes necessary, the hauling yoke of the caisson is temporarily disconnected, the several valves of the water-chamber are closed, and as the water rises and the displacement of the caisson becomes equal to its floating weight, the vessel will float and can be removed, as a ship or pontoon, to the outer meeting face of the dock entrance above the apron. The valves are then opened and the caisson is sunk in the outer track.

The caisson framing is composed of steel angle-bars and plates, as follows :—

The frames are of angle-bars 4 inches by 3 inches by $\frac{1}{2}$ inch, spaced 18 inches apart, and vertical and bottom corner angles 4 inches by 4 inches by $\frac{1}{2}$ inch. The frames are cut at the water-tight deck, and are connected to same by plate brackets 22 inches by 18 inches by $\frac{7}{20}$ inch above and below. The upper brackets are connected to the deck by an angle 3 inches by 3 inches by $\frac{7}{20}$ inch, one on every frame. The alternate frames at the bottom are connected to the floor-plates, and the outer frames are connected to plate brackets 18 inches by 18 inches by $\frac{1}{2}$ inch, and 3 inches by 3 inches by $\frac{1}{2}$ inch angles on bottom. All frames extend from the top of the bottom corner angles to the underside of the water-tight deck, and from the top of the water-tight deck stringer angle to the underside of the rail angle. The vertical corner angles extend, in one length, from the bottom of the caisson to about 12 inches above the water-tight deck, and a short length from thence to the rail angle.

The bottom floors are 30 inches deep by $\frac{1}{2}$ inch thick, spaced 3 feet apart, the bottom angle being 3 inches by 3 inches by $\frac{1}{2}$ inch, and the top angle 3 inches by 3 inches by $\frac{7}{20}$ inch. The bottom angle extends from the inner edge to the inner edge of the bottom corner angles, the top angles extending from heel to heel of frames.

The plating of the water-tight deck floor is $\frac{7}{16}$ inch, with all butts and seams overlapped, seams single, and butts double-riveted, $\frac{3}{4}$-inch diameter rivets throughout, spaced $2\frac{5}{8}$ inches in butts, $5\frac{1}{4}$ inches in beams, and 3 inches in seams. The stringer angle on the upper side of the deck is 3 inches by 3 inches by $\frac{1}{2}$ inch, riveted to the shell with $\frac{3}{4}$-inch rivets, spaced 3 inches apart. The beams carrying the deck are 5 inches by 3 inches by $\frac{1}{2}$ inch angles, spaced 18 inches apart, and secured to the frames with $\frac{7}{20}$-inch plate knees. The cross-beams under the water-tight deck are 4 inches by 4 inches by $\frac{1}{2}$ inch angles, and those above the water-tight deck are 3 inches by 3 inches by $\frac{1}{2}$ inch, spaced 3 feet apart, and secured to the frames by plate knees, $\frac{7}{20}$ inch in thickness. The vertical supports below and above the water-tight deck are $3\frac{1}{2}$ inches by $3\frac{1}{2}$ inches by $\frac{1}{2}$ inch, and 3 inches by 3 inches by

$\frac{1}{2}$ inch, respectively, secured to the beams with one rivet in each. The stringers below the water-tight deck are of plates, 18 inches by $\frac{7}{20}$ inch, and angles $3\frac{1}{2}$ inches by $3\frac{1}{2}$ inches by $\frac{7}{20}$ inch, and those above that deck are 15 inches by $\frac{7}{20}$ inch and 21 inches by $\frac{7}{20}$ inch, the angles being $3\frac{1}{2}$ inches by $3\frac{1}{2}$ inches by $\frac{7}{20}$ inch. All butts are overlapped and are single riveted, $\frac{3}{4}$-inch rivets throughout, spaced $3\frac{1}{2}$ inches apart in butts, and $4\frac{1}{2}$ inches in angles.

The shell plating of the caisson is as follows:—

| | |
|---|---|
| 1st strake of sides and ends . . . | $\frac{11}{20}$ inch thick. |
| 2nd ,, ,, ,, ,, ,, . . . | $\frac{21}{40}$ inch ,, |
| 3rd and 4th ,, ,, ,, ,, ,, . . . | $\frac{10}{20}$ inch ,, |
| 5th and 6th ,, ,, ,, ,, ,, . . . | $\frac{9}{20}$ inch ,, |
| 7th ,, ,, ,, ,, ,, . . . | $\frac{25}{40}$ inch ,, |
| 8th ,, ,, ,, ,, ,, . . . | $\frac{7}{20}$ inch ,, |
| 9th ,, ,, ,, ,, ,, . . . | $\frac{6}{20}$ inch ,, |

The bottom plates are $\frac{11}{20}$ inch in thickness.

All plates are lapped, the seams being single, and the lap-butts double-riveted throughout. The spacing of the rivets in the butts is $2\frac{5}{8}$ inches, in the frames $5\frac{1}{4}$ inches, and 3 inches in the seams. The bottom plating in the way of the keelsons have flush butts with double-riveted straps inside, and all bottom riveting is knocked down from the inside. Four adjusting boxes, with screw and links of steel, are fitted to the ends of the caisson, two at either end. They are worked from the upper stringer by means of a key. These are for bearing the caisson up against the granite meeting faces.

The two endless hauling chains, which are supported by twenty cast-iron rollers along each side of the caisson recess, are $1\frac{1}{2}$ inches short link. The driving-shafts are $8\frac{3}{4}$ inches and 8 inches in diameter, respectively, each having a boss for a chain-pulley, and cast-iron couplings are turned and keyed on the shafts, and are fitted with turned bolts and nuts.

The contract for the caisson was entrusted to Messrs. Hanna, Donald and Wilson, of Paisley, the total cost amounting to £15,454 12*s.* 7*d.*

*Caisson Hauling Plant.*—The hydraulic plant, Fig. 10, Plate 33, which has been provided for the hauling of the caisson gate either into or out of the recess in the process of opening or closing the dock, is housed in an underground chamber on the eastward side of the dock entrance. It is capable of being actuated either by hydraulic power, or, in case of emergency, by manual power. The machinery is coupled up to a driving-shaft in the caisson recess, which shaft is provided with two sprocket wheels engaging a pair of endless chains that are attached to the hauling yoke fitted on the end of the caisson; the chains are supported on a series of rollers placed along both sides of the caisson recess 5 feet and 6 feet 9 inches, respectively, below the surface of the quay. When this machinery is being worked by hydraulic power under the accumulator pressure before referred to, it is capable of exerting a hauling force of 24 tons on the caisson at a speed of $14\frac{1}{2}$ feet per minute, or 12 tons at a speed of 29 feet per minute.

The hydraulic engine is of the horizontal type mounted on a cast-iron bed-plate, and has three cylinders fitted with rams, crossheads, and connecting-rods acting directly on the crankshaft. The cylinders and rams are of gun-metal, and the crossheads are of steel with gun-metal faces working in guides on the bed-plate. The connecting-rods are of steel with adjustable gun-metal brasses at both ends. The forged steel crankshaft works in bearings carried by the bed-plate and fitted with adjustable gun-metal brasses. The working valves are of gun-metal of the slide pattern, actuated by eccentrics on the crankshaft and fitted with reversing slides which actuate, through a shaft and levers, by a hydraulic cylinder of gun-metal. The spur-gear is provided with two sets of gearing wheels to give the two powers above referred to, a clutch being fitted for putting either set of gear into or out of gear; the clutch is actuated by a portable hand-lever from above the quay level. The hand gear consists of a head with sockets for hand-spikes and bevel gearing driving on to the intermediate shaft. All the gearing is of steel with machine-cut teeth, and mounted on steel-shafts working in bearings having adjustable gun-metal brasses. A brass-lined stuffing-box and gland are placed at the

connection between the hauling plant and the main shaft in the caisson recess, for the purpose of preventing the entry of the tidal water from the latter into the machinery chamber. The motion of the hydraulic engine is controlled by a gun-metal valve operated by a portable hand-lever from above the quay level; the reversing cylinder being operated by means of the same valve and lever.

This contract was entrusted to Messrs. Sir W. G. Armstrong, Whitworth and Co., Ltd., of Newcastle-on-Tyne, and cost £921 19*s.* 2*d.*

*Penstock Doors for Culverts.*—There are nine penstock sluice-doors on the various dock culverts, namely, four 9 feet by 9 feet, one 7 feet 6 inches by 6 feet, and four 7 feet 6 inches by 5 feet, clear openings, Fig. 11, Plate 34. These doors are operated by hydraulic power supplied at a pressure of 750 lb. per square inch, and are arranged that in case of a failure of the water-pressure the sluices can be opened and closed by hand power. The doors are double faced, and are constructed of greenheart beams. For the 9 feet by 9 feet doors there are ten beams, each 10 feet 8 inches long by 13 inches deep and 18 inches wide. Each door has a massive cast-iron beam at the top, having eyes to receive the pin which connects the sluice-rods to the door.

The greenheart and cast-iron beams are all carefully dressed and fitted to each other, and are secured by seven $2\frac{3}{4}$-inch bolts passing right through the full depth of the door, and are fitted with cast-iron countersunk washers at the bottom end, the whole being made perfectly water-tight. The hydraulic cylinder for operating the door is of cast-iron, $20\frac{3}{4}$ inches inside diameter, bored out through its entire length; the thickness of the metal of the cylinder is $3\frac{3}{8}$ inches, and strong brackets are cast on and rest on two mild-steel box-girders 18 inches by 21 inches, built into the wall of the chamber; the cylinder is securely fixed to the girders by four $3\frac{1}{2}$-inch bolts. It is also provided with the necessary inlets, air, and frost cocks.

The piston is of gun-metal, in three pieces, having two leather cups, and is secured to the piston-rod by a gun-metal nut. The piston-rod is fitted with a strong cast-iron crosshead, having eyes for the sluice-rods, which extend from the crosshead to the door and which are secured by pins at each end. The sluices are operated by a 1-inch double-ported slide-valve, carried on a cast-iron bracket, which is bolted to the masonry at the top of the chamber, this being fitted with a 1-inch stop-valve, a tee-piece bend and a connection for hand power; and all the necessary piping between valve and cylinder is supplied.

The valves are operated by a hand-lever made so as to be removable and working through a slot in the chamber covers. The door, 7 feet 6 inches by 6 feet in the clear, is constructed on the same lines as above; the greenheart beams being eight in number, 7 feet long $12\frac{3}{4}$ inches deep by 14 inches wide, are bolted together by five $2\frac{3}{8}$-inch tie-bolts. The cylinder for operating same is $15\frac{1}{8}$ inches diameter and $2\frac{1}{2}$-inch metal; in this case the cylinder has strong brackets or feet cast on for fixing to the granite wall of the chamber, which is done by four 2-inch bolts; and in addition it is secured to the opposite wall of the chamber by strong bolts and heavy cast-iron brackets built into the wall. Suitable mild-steel sluice-rods extend from the piston-rod to the door, and where necessary the rods are made in suitable lengths, jointed together by strong socket couplings and cotters and guided by cast-iron guides fixed to the wall. The four 7 feet 6 inches by 5 feet doors are also constructed as above, the greenheart beams being eight in number, 6 feet long, $12\frac{3}{4}$ inches deep and 13 inches wide, and they are bolted together by five 2-inch tie-bolts.

The cylinders are 10 inches, $11\frac{1}{2}$ inches and $12\frac{1}{4}$ inches diameter, respectively, according to the head or depth under which the sluices require to work, the thickness of metal being $1\frac{5}{8}$ inches, $1\frac{7}{8}$ inches, and 2 inches. The operating slide-valves are $\frac{3}{4}$ inch diameter.

The chamber-covers for all the sluices are of cast-iron chequered on the top, and placed in a check made in the granite masonry. Slots and eyes are founded in the plates through which to reach

the working valves, so that the sluices can be operated without removing the plates. All those parts which are subjected to hydraulic pressure are designed for a working pressure of 750 lb. per square inch, and were tested to a pressure of 2,500 lb. per square inch.

The contract for the penstock doors and hydraulic operating gear was entrusted to Messrs. Glenfield and Kennedy, Ltd., of Kilmarnock, the cost altogether amounting to £2,929 14*s.* 2*d.*

*Pumping Station Buildings.*—The contract for the handsome pile of buildings, which has been erected to contain the pumping and hydraulic plant and boilers, together with the factory chimney, 180 feet in height above the ground line, was entrusted to Messrs. McLaughlin and Harvey, Ltd., of Belfast, and cost £9,521 18*s.* 8*d.*; which sum includes only the buildings of the engine and boiler-house, hydraulic room, and chimney, above ground line.

The Paper is illustrated by Plates 27 to 34 and 2 Figs. in the letterpress.

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## *Discussion.*

The PRESIDENT, in moving a vote of thanks to the author for his interesting account of an important work, said the Members had been given an illustration of the advantages they possessed in coming to Belfast. Belfast was famous for things which it did in the biggest manner. It had the biggest shipbuilding yard; it had the biggest dock; it had the biggest flax works, and, he believed, the biggest distillery. On the following day the Members would have the opportunity of seeing the machinery which the author had described. The Paper was not one which lent itself to prolonged discussion, because it was simply an account of an interesting work, in connection with which the area for difference of opinion was not great.

The resolution of thanks was carried by acclamation.

Mr. Charles J. Hobbs, in opening the discussion, said that personally, as a manufacturer and as one interested in hydraulic machinery, he congratulated the author on the choice of the contractors whose names he gave in the Paper and who were entrusted to carry out the work. He also congratulated the author on being able to purchase his machinery at a very low price. His own company was not successful in obtaining an order for any of the work; but he remembered at the time that trade was not as good as it was at present, and for that reason the author was fortunate in the cheap price at which he was able to place the work. He was also very pleased to find that the author had adopted hydraulic power. He might mention that at the present time another large dock was being built, known as the Hull Joint Dock, under the superintendence of the joint engineers of the North Eastern Railway and the Hull and Barnsley Railway, and hydraulic power was also being used there for the same purposes, namely, for the larger capstans, the penstock machinery and the dock-gate machinery, which answered the same purpose as the caisson machinery in the author's case, while electricity was being used for some of the other details.

He noticed on the previous day when going along the river that a great deal of steam-power was used for the manipulation of coal along the dock side; he referred to the coal wharves nearly opposite Messrs. Harland and Wolff's Works; and as he believed there was some prospect of these cranes being altered, he hoped that the authorities would adopt the idea of using hydraulic power. He did not wish to start a discussion on hydraulics versus electricity, although this was a typical case where hydraulic power might be used with advantage, inasmuch as the height of lift and load to be raised were constant, and these were advantages telling in favour of hydraulic power.

The author had referred in a very clear manner to the reason for the use of compound engines non-condensing. There were, however, one or two points he would like to raise in connection with the design of the engines which he could not quite follow, because the illustration on Plate 29 was drawn to a very small scale. He

(Mr. Charles J. Hobbs.)

gathered from the drawing that the pump-bodies were cast in one with the front vertical engine frames. It had occurred to him that that was liable to lead to trouble. In the event of the pump-body breaking, which sometimes occurred, it meant that the whole of the front frame would have to be removed and renewed. He was not quite sure whether that was the case or not; but the members would have an opportunity of seeing the machinery in question on the following day. It looked as if the pump-body and the delivery-pipe and the suction and delivery valve-bodies were all cast in one piece with the front columns.

He noticed that the author referred on page 666 to the Admiralty capstans, which had 16 tons capacity. It was quite correct that that was the largest size the Admiralty had adopted up to the present, the last ones ordered being for Chatham, and these his firm were manufacturing at Chester at the present time. In the latter design the cylinders were not arranged with three-throw cranks, as shown in the Fig., but with two cylinders, double-acting, on cranks at right angles. In concluding, he hoped Mr. Kelly could see his way to separate some of the costs given in the Paper for some of the details, as this would no doubt be useful for future reference.

Mr. T. F. Shillington said that, as a Member of the Belfast Harbour Board, he felt it difficult to speak altogether impartially in regard to a work just recently carried out by the Harbour Commissioners themselves. In the first place, he wished to acknowledge the author's recognition of the fact that the Harbour Commissioners had at all times shown a desire to further the shipbuilding industry and everything connected with the progress of the port; but if he were to go on in that line those present would think that the Harbour Commissioners were more a society for mutual admiration than for looking after their legitimate business. In large works of the kind under discussion he was quite sure that, notwithstanding the best intentions of those who had to carry them out, there were some things which on looking at afterwards, one might like to have carried out differently, and that if they had to do the same thing again they perhaps would

try to do better. But it was not probable that Belfast would often have a chance of spending £350,000 on a dock, so that the present Commissioners were not likely to have an opportunity of making use of the experience thus gained.

The engineering difficulties had been enormous. The foundations, which he supposed were more a matter for the consideration of Civil Engineers than for Mechanical Engineers, caused great trouble owing to running sand, and anyone who had had experience in this subject knew the great difficulty of dealing with sand after it had got in motion. If the trouble was overcome in one place, it was always liable to crop out in another, and that caused the author and all connected with his department an immense amount of labour and anxiety.

He was glad to say that all the obstacles had been surmounted, and that the dock as at present completed was a monument that would last as long as Belfast was in existence. The only prospect of alteration was in regard to the possible lengthening of the dock, as shipbuilders were likely to go ahead in the construction of even longer vessels. They had already heard of bigger things, not only in this country but elsewhere, and it was quite possible that in the future they would have to lengthen the dock. In that case they might have to face again some of the difficulties with which they had had to contend in the past.

Mr. WILLIAM H. PATCHELL (Member of Council) said that there were one or two points on the engineering side of the question that he would like to ask. He congratulated the author on the very able way in which he had read the abstract of his Paper, and the short and pithy way in which he had put it and additional details before the Members. He had intended to ask why he had adopted non-condensing engines, but this question was partly explained by the striking fact, now mentioned by the author, that the pumps were only used for 104 hours per annum. If condensing engines had been installed, the boiler plant could have been cut down considerably. He hoped the author would say a little more about the pumps, as it would be of interest to know whether they

(Mr. William H. Patchell.)

were self-regulating or whether they were regulated by hand. The quantity to be discharged was given as 21,000,000 gallons in 100 minutes. This gave an average discharge of 70,000 gallons per pump per minute. The maximum lift was 33 feet, and it was evident that the 740 h.p. pumping engines could not handle 70,000 gallons against a 33-foot head, as that would mean an over-all efficiency of over 94 per cent., so some method of regulating was necessary; and he would be glad if the author could supplement his Paper with particulars of the pump duties and efficiencies under the regulating conditions for the maximum and minimum lifts.

He noticed also that the dock, which was of the enormous length of 850 feet, was not divided, but from the plan he noticed the dock alongside appeared to be divided by two inner doors. Possibly the adoption of a middle door to the dock would enable it to be used more often in the year than appeared to be the case at present.

With regard to the question of hydraulic capstans, a hauling stress of 30 tons at 30 feet a minute was mentioned as the capacity of the most powerful capstans ever yet employed. This only meant 900 foot-tons per minute or about 60 effective h.p. He dared not say anything disrespectful about the efficiency of hydraulic machinery in the presence of the President, and would leave the figure of the i.h.p. of the haulage engine subject to any correction for efficiency that the President insisted on. He suggested, however, that a 60 h.p. haulage engine was not by any means a large one.

He would also be glad if the author would give a few more details about his steam-engine plant, because apparently the water per i.h.p. per hour worked out very nearly at 30 lb. per h.p.-hour. Possibly he worked, as they did in America, for an over-all commercial efficiency in a plant which was of a very robust type and without refinements, and which could be looked after practically by unskilled men. He thought, however, that in Belfast it would have been quite easy to obtain skilled men, and that some little refinements as regards fuel might have been worthy of consideration, because fuel had to be imported into Ireland.

He was glad to notice that the author had adopted the British Engineering Standard specification for steam-pipe flanges, particularly as regards the thick flange. Dr. Maw was the Chairman of the Pipe Flanges Committee, and the question of the thickness of flange was very carefully considered. Personally he stood out on the Committee for a thick flange, because he believed in heavy steam-pipe flanges, as they made for sound work and saved time in erection. The joint-rings could be dropped in between the bolts after the pipes had been erected, and a very strong joint was obtained. He was rather astonished lately to see in a Home Office Memorandum on Steam Boilers—a statement practically prohibiting the joint inside the bolt circle; it was, without reserve, held up as an object of execration. He regretted that should be done in a Government publication.

Mr. Walter Dixon said the Paper was one dealing with facts, in connection with which two or three points stood out prominently. In the first place, the author referred to the new graving dock at Belfast as being the largest in the kingdom. He happened to come from a city—Glasgow—which was contemplating at the present moment making an even larger dock than the one that had been built at Belfast. As had been pointed out by the author, it was necessary for engineers to bear in mind the growing necessities not only of the merchant service, but also of the Admiralty; and therefore a Paper containing such information as that embodied in Mr. Kelly's Paper called for special consideration.

The first point that struck him was that the dock was started seven years ago, and naturally during that period a considerable change had been taking place in engineering practice. His mind went back from the present Meeting to what the Members saw in Switzerland in the previous year. While the plant which had been installed in the Belfast Dock was no doubt excellent, another phase of things has arisen since seven years ago. He did not wish for a moment to enter into the question of electricity versus hydraulics, nor the advent of the Diesel engine, but he thought any plant installed in a dock similar to that dealt with in the Paper would

(Mr. Walter Dixon.)
naturally call for reconsideration in view of the many things that had happened in the past seven years.

The author enlarged upon another point in his remarks which personally he thought were most interesting. He had mentioned that the dock cost £350,000, but it was rather startling to mechanical engineers to hear that the machinery cost only about 10 per cent. of the total sum expended. Leaving out the cost of the caissons, he found that of the £350,000 only about £30,000 was spent on the machinery. As one of the speakers had already pointed out, he did not think the machinery portion should have been cramped. If £300,000 was expended on the dead plant, the question of the cost of the machinery or the actuating part of the undertaking was a mere detail. Of course the question of working cost came in; but when it was borne in mind that steam had to be kept up in the plant more or less constantly, he thought it would be found that engineers in designing future docks of a similar kind would hesitate very greatly before installing steam plant which must be kept going constantly.

He wished to refer to a point mentioned by one of the previous speakers, namely, the subsidiary plant. When the small amount of power involved was taken into consideration, namely, 150 h.p. for the hydraulic plant and 60 to 100 h.p. for the capstans, he thought it was a little surprising in a city like Belfast, with its electricity supply, that they should have gone to a large expense in installing primary plant when presumably the city supply might have given them all the smaller powers, which were only used, as the author said, from 50 to 100 hours a year.

The PRESIDENT said he was sorry that the author had been called away to attend a Meeting of the Harbour Commissioners, but a proof of the discussion would be communicated to him, so that he would be able to reply in writing.

Mr. W. REDFERN KELLY wrote that, with reference to Mr. Hobbs' observations (page 676) as to the hydraulic capstans provided at Belfast, these were designed with three-throw cranks

for balancing, and gave a fairly even turning-moment, the engines being single-acting.

In regard to Mr. Patchell's remarks (page 677) on the pumping plant, the engines were not self-regulating, but were controlled by a hand-wheel and screws connected with the cut-off valves on the back of the slide-valves of the high-pressure cylinder, and were thus adjustable while the engines were working. As to the power of the pumping plant, the average discharge for each pump was 70,000 gallons per minute; the maximum lift was 33 feet, but it was not claimed that 740 i.h.p. could deal with 70,000 gallons per minute against a 33-foot head, as could be seen by reference to the diagram, Fig. 12 (page 682), which showed the efficiency at varying heads and during equal periods of the official test. Mr. Patchell's suggestion that had condensing engines been installed the boiler plant could be cut down considerably, was, in a sense, quite correct, but if done it would mean a considerable extra expenditure for condensing plant and loss of valuable space which could ill be afforded.

The question of dividing the new Graving Dock into two compartments was duly considered when the dock was designed, but was regarded as not being desirable. The very serious outlay which would be entailed in the construction of a new intermediate entrance with its caisson gate, recess, culverts, penstocks, etc., could not at all be justified in the present instance. The new dock would be mainly used for vessels of very large type, such as the S.S. "Olympic," etc., as the adjoining dock would, for all vessels which it could accommodate, be regarded for a considerable time to come as the popular dock for general use, for many good and sufficient reasons.

With regard to the consumption of water for the engines, this amounted to about 22 lb. per i.h.p. per hour, and not 30 lb. as suggested by Mr. Patchell. This also would be apparent by reference to the diagram, Fig. 12 (page 682). The plant adopted was not of the robust American type suggested by Mr. Patchell, but had such refinements as a Royle's feed-water heater, superheated steam, cylinders completely jacketed, slide-valves of Martin and

(Mr. W. Redfern Kelly.)

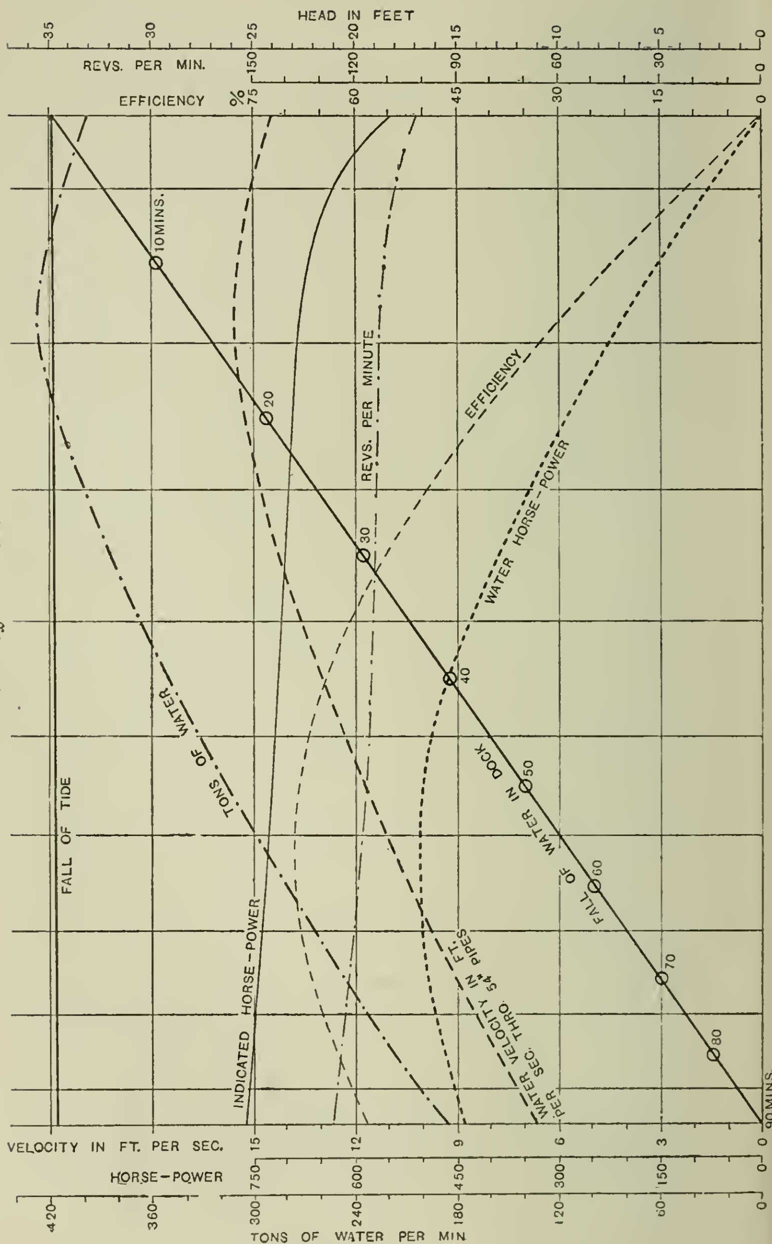


Fig. 12.—*Official Tests.*

Andrews' balanced type, with cut-off valves in the high-pressure cylinders and adjustable while working, positive lubrication to cylinder and valve-faces, etc., and the plant was of course in charge of skilled men. As to the consumption of fuel, it would be seen by reference to the diagram that about $4\frac{3}{4}$ tons of coal only were consumed within a period of ninety minutes—the time occupied in pumping out 19 million gallons of water from the dock.

With reference to Mr. Dixon's observations (page 680) as to the small relative cost of the mechanical plant for the new dock, the author would like to state that this plant had not in any sense of the term been "cramped." It was, in every respect, thoroughly efficient, and more than equal to any demands that might be made upon it at any time.

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*Particulars relating to the Curves on Fig.* 12.

Centrifugal Pumps, 54 in. diameter.
Impeller, 7 feet 6 inches diameter.
Vertical Compound Non-Condensing Engines, cylinder 22 inches by 38 inches diameter by 20-inch stroke.
Steam-Pressure, 160 lb. per sq. inch.
Superheat, 100° F.
Average i.h.p., 715.
Average Water h.p., 385.
Average Efficiency, 54 per cent.
Steam Consumption, 22 lb. per h.p.-hour.

Quantity of Water pumped, 19,000,000 gallons.
Time of Pumping, 90 minutes.
Fall of Tide during Test, 4 inches.
Boilers of Babcock-Wilcox Type.
Coal consumed during Test, 4 tons 15 cwt.
Feed-Water to Boilers during Test by meter, 6,960 gallons.
Three Boilers used during Test.
One Boiler kept in reserve.
Blowing of Escape Valves = 1 Valve.
Blowing 9 min. 30 sec. = 2,000 lb. water.

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# THE EVOLUTION OF THE FLAX SPINNING SPINDLE.

By JOHN HORNER, of Belfast.

The author need make no apology to the members of this Institution for going back to a past which, to the minds of the unpractical, should be long forgotten. It is from no purely archæological standpoint that he resurrects the spindle of the ancients. A desire to better any mechanical device should be accompanied with a knowledge of what led up to its present state and to trace by such knowledge the reasonings employed in its development. The inventive faculty is greatly stimulated by an understanding of all the motives which influenced men's minds in the past. Often the very principles which underlie some mechanical construction are forgotten, simply because the problem has long since been solved, but before this solution the difficulties and failures caused a thoughtful study into their origin, and thus of necessity the fundamental laws which govern a process are brought to the front.

Amongst some primitive people to-day fibres are twisted together merely by the action of the index fingers and the thumbs, and then plaited to form a cloth, or used as bow-strings or fishing tackle by again twisting the yarn produced into several plies.

When a knowledge of the length, combined with strength, that short fibres could be made to assume when thus united together became known to man, he probably used such naturally provided means to spin his yarn. When and how the spindle came we have no means of ascertaining; it may have been evolved from something cruder still—a twirling stone for example—but if it jumped into existence at once, it might be considered perhaps the greatest forward bound in the whole realm of spinning invention. We have intimate knowledge of its use a long while before authentic history began. It is found amongst the remains of neolithic man, and the lake dwellers of Switzerland. It was handed down to Egypt, for it is found in the tombs at Thebes, also amongst the ruins of Babylon, and in the Inca groves of Lima (*a*), Fig. 1, Plate 35.

Simple in construction and decidedly effective in use, the spindle in its primitive form has descended from remote prehistoric times to the present day, and is found in actual practical use amongst the various tribes in Africa, the Mongols in the Far East, throughout our Indian Empire, in Persia and over the European Continent. The spindle is usually composed of two parts—the actual spindle itself and the whirl or whorl which adds to its momentum. The former is of wood, commonly notched at the top to receive the end of the spun yarn. The latter is of wood, stone, bone, clay or other convenient substance having sufficient weight to maintain the desired momentum. The word "whirl" is sufficiently expressive. "Whorl," simply another form of "whirl," is derived from the Old English word "whorvil," meaning the whirl of the spindle, and from this the botanical name is derived. When the spindle was afterwards fixed in bearings and driven by a band, a sheave or small grooved pulley was fastened about where the whorl was originally placed. This has since received the name of "wharve," perhaps from the association of sound, or it may come from an older Anglo-Saxon derivation. The word "wharve," although always used to designate this grooved pulley in modern spinning frames, does not appear in this sense as a dictionary word. The spindle with the whirl or whorl attached is given rotation by briskly rolling it with the palm of the hand against the thigh,

a practice universal in Southern Italy, or, more generally, by spinning it between the thumb and index finger. The fibres are then evenly drawn from the distaff and twisted into yarn by the revolving and descending spindle, which, when it reaches the ground, may still continue to perform its function. When at rest the spun yarn is wound on the spindle, the last spun portion inserted in the notch and the operation continued. It requires very accurate and precise manipulation to determine the diminution in feed to suit the diminishing speed of the spindle, and only by early training can this be accomplished. It is generally believed that the art of spinning is one that is only suited to the sensitive touch of the female hand. In the reign of Frederick the Great all male farm-hands in Silesia were required to learn spinning, so that they might utilize the otherwise unprofitable dark winter evenings, and at this they became very adept.

An ancient Peruvian spindle is shown in (*a*) Fig. 1, Plate 35. The following illustrate some types in use to-day photographed from actual specimens in the Belfast City Museum. One from Egypt is shown at (*b*), and does not vary from many of those used in that country centuries ago. (*c*) is South Italian; hemp is the fibre universally spun there. South of Rome spinning wheels do not exist; some fifty years ago they were introduced, but the sedentary occupation did not appeal to the lively nature of the Southern Italian. Everywhere one sees in South Italy the production of yarns by this primitive method. With the distaff stuck in the breast or girdle, the women spin as they walk along or stand in the markets or chat with each other in the house or outside. Within the shadow of the most modern and up-to-date spinning concerns this ancient method is quite alive. A Russian hemp spindle is shown at (*f*), and (*d*) is one of smaller build for flax; the distaff (*e*) has often hemp bound to one side and flax to the other, and may be used by two or more women together. A similar arrangement of stand distaff is shown at (*g*), with the hand spindle used to-day in Germany, particularly in Posen. It is a curious fact that, in Prussian Silesia, the chief linen-producing centre of what is now the German Empire, this ancient method of spinning continued

generally until after the introduction of spinning machinery; passing over and ignoring the intermediate stage of the spinning wheel, a less twisted, and consequently a softer, yarn was produced, which gave character to the Silesian woven linen, and made it more accessible to bleaching agencies. Other illustrations of European spindles such as are used in South France, North Italy, Switzerland, Spain, Portugal, etc., are unnecessary; they are very similar to the German type.

A spindle from the Congo is shown at (*h*), Plate 35, the whorl being made from cassava root; and (*i*) shows one from Nigeria; the whorl is of pottery ornamented in a manner closely resembling that from Peru (*a*), Fig. 1. One from Madeira (*k*) has a groove in the top in which the spun yarn is fastened; this spindle has no separate whorl, the body of the spindle itself being thickened to serve the same purpose.

Fig. 2, Plate 36, shows one from India, and has a small copper coin in use as a whorl. In the illustration, besides the small distaff on which the cotton is wound, is seen a portion of a cocoanut, in the concave side of which the spindle was sometimes revolved as in a bearing. An illustration is given of this along with descriptive matter in a volume written in 1867 by J. Forbes Watson on the textile manufactures and the costumes of the people of India. The extraordinary fineness of the yarn produced for the Decca fabrics is almost beyond belief. The yarn spun was so delicate that it could not bear the strain of a tiny wooden spindle weighted with a minute piece of clay for a whorl, and consequently, to relieve the strain, the spindle was required to revolve in the hollow of a cocoanut or the concave side of a shell. In the authority above mentioned there appears a quotation from a work descriptive of the manufactures of Decca, written by James Taylor in 1851, in which the following appears: "A skein, which a native weaver measured in my presence in 1846 and which was afterwards carefully weighed, proved to be in the proportion of upwards of 250 miles to a pound of cotton."

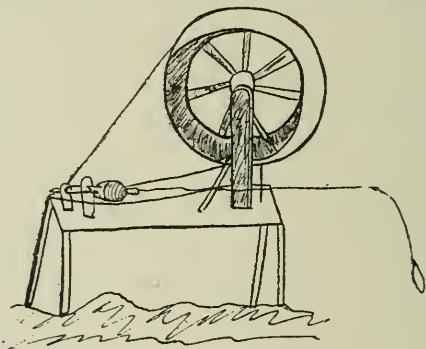
Passing from the primitive spindle revolved as described, we arrive at the first known spinning-wheel, that of India, Fig. 3,

Plate 36. This wheel has been in use in that country, according to the vague phrase so commonly used, from time immemorial. This indicates a very great antiquity. The evolution was a simple one; it consisted in placing the spindle horizontally in bearings and revolving it by means of a cord or band over a driven wheel to a grooved pulley or wharve on the spindle. This wheel, it will be seen, is unsupported by legs, and is thus suited to the attitude of the Oriental spinner who sits or kneels on the ground while at work. Although a more continuous spindle-speed is arrived at, still there is a disadvantage. One hand is constantly employed in revolving the driving wheel, which can be maintained at even speed; this leaves but the fingers of the other hand for the delicate operation of attenuating the fibres, which can be done with much greater accuracy and precision when the fingers of both hands are employed; and no doubt for this reason the fine Decca muslin yarn continued to be spun by the older method. In China they attempted to get over this difficulty by producing a wheel with a treadle motion, Fig. 4, Plate 36. This mode of converting a reciprocating into a rotary motion is one which, the author ventures to say, may possibly be unknown to the members of this Institution, and therefore unrecognized in modern mechanical practice. It will be better understood from the actual wheel which will be on exhibition at the Meeting; *a* is the driven wheel revolving three spindles, *b*; *c* is a lever with fulcrum at *d*, and entering one of the spokes of the wheel at *e*. The wheel is inclined at an angle so that the lever remains in position whether directed up or down; with a foot on either side of the fulcrum the lever is swayed upwards and downwards and round with the wheel. This ingenious device requires one operator to revolve the wheel, thus placing three spindles at the disposal of as many spinners, each with both hands at liberty.

China and Japan, as well as India, use in the spinning-wheels a simple spindle driven by a band. The earliest intimation we have of this type of wheel in Europe is contained in illuminations in a fourteenth century manuscript in the British Museum Library. One of these is reproduced in Fig. 6 (page 690). The wheel in its movements is the same as that of India, but is mounted on legs to suit the

operator, who, in the illumination, is depicted standing. This class of wheel, as will be seen later, survived in England for cotton spinning until the introduction of mechanical frames for that fibre. It is in use to-day in many parts of Scotland and Ireland for spinning wool, and until very recently—perhaps still is—in Holland for producing hemp yarns. In all these wheels, as in the simple spindle, the two operations necessary, namely, the actual spinning and the winding of the spun yarn on the spindle body, were performed at different times, namely, the yarn was spun about two

FIG. 6.—*Simple Spindle driven by a Band.*
(From Fourteenth Century MS. in British Museum.)



yards or so in length and was then allowed to wind on the revolving spindle.

The next step in advance was not one of simple evolution, but one which entailed considerable thought and constructive ability. The problem to be solved was how to spin and wind conjointly. To a woodcarver called Johann Jürgen, of Wattenbüttel in Brunswick, has been ascribed the honour of inventing an ingenious and altogether novel device for producing this effect, and the author thinks in all justice it may be said that his invention was independent; but a great master mind had solved the problem before him. Leonardo da Vinci, who died in 1519, some years before Jürgen brought out his invention, left behind him a vast

number of sketches which probably were never in any way made public until they were presented to the Ambrosian Library in Milan in 1639. One of these sketches, Fig. 7, Plate 37, is of a spinning wheel and embodies principles of absolute originality. The explanatory matter attached to this sketch, written with the left hand in Leonardo's usual manner from right to left, is greatly abbreviated and not easily understood except by a savant. Through the medium of a friend in Milan, the author has had this matter put into modern Italian by one of the Ambrosian monks who can read Leonardo's characters, and it is now attached to the drawing converted into English.

Before dealing with Leonardo's description of his drawing, the principle of the spindle, flyer, and bobbin involved in it needs some brief explanation. The indications in letters on the drawing are Leonardo's; those in figures have been inserted by the present author. In the original method the actual spinning was carried out from the point of the spindle. In Leonardo's, which combines spinning and winding at the same time, the spinning is effected by means of the flyer 1 attached to spindle 2, the unspun material being passed through a hole in the spindle and conducted along one of the flyer legs; when spun, it is automatically and immediately wound on the bobbin 3. In order to effect this winding the flyer and the bobbin must run at different speeds, otherwise there would be no take-up. If the flyer runs quicker we have flyer-lead, if the bobbin we have bobbin-lead; in either case the effect, so far as winding the yarn on the bobbin is concerned, is the same. The larger wheel 4 drives the spindle 2, to which the flyer 1 is fastened, at a quicker speed than the smaller wheel 5 drives the bobbin 3. The pegs placed between the larger wheel and the band for tightening the latter give a harder twist to the yarn; one or more of these could evidently be inserted at pleasure.

To turn to Leonardo's own descriptive matter, where he begins "The wheel S" we have a motion for automatically building or winding the yarn on the bobbin evenly. This idea never occurred to anyone during the long period of wheel-spinning and was introduced, evidently quite independently, into modern cotton

machinery some 250 years later. This arrangement of Leonardo's for building the yarn on the bobbin evenly as it is spun gets its first motion by a worm 6 on the extended driving shaft 7 gearing into a worm-wheel A G. On the surface of the latter are a series of teeth 8 working into two incomplete lantern wheels 9 at top and bottom of S; this is shown more distinctly in the detail Fig. 8, Plate 37. Each of these lantern wheels has four teeth placed in opposite directions; between these two wheels is a forked lever 10, not fully extended in the detail Fig. 8, which traverses by the motion of the gearing the square shaft of the spindle 2 to and fro, and thus builds the spun yarn upon the bobbin. It is not probable that these original ideas were ever put into actual practice. After the problem was solved on paper, it was put away along with many hundreds of other mechanical devices. The spindle, flyer, and bobbin introduced by Jürgen in 1530, and, what is more to the point, made of practical utility, embodied the spindle, flyer, and bobbin substantially as Leonardo had them. Some authorities state that these were introduced first on what is known as the Saxon wheel and that Jürgen was only responsible for a treadle motion. The authority in support of Jürgen is Professor Hugo von Rettich in his "Spinnrad-Typen." Jürgen had no building motion, from which fact one would draw the evident conclusion that he had not seen Leonardo's sketch. To arrive, however, at this a series of wire hooks were inserted in each leg of the flyer, Fig. 9, Plate 38 (parts of an Irish spinning wheel); when a certain portion of yarn was wound it was permitted to run through the neighbouring hook and so on until the bobbin was fully, but by no means evenly, wound. Fig. 10 shows another method: the flyer legs are pierced with a series of holes into which a piece of naturally bent wood or a piece of bent wire is inserted and moved from hole to hole as occasion requires.

Fig. 11 is an iron flyer in which slots are cut at angles to serve the same purpose; the angular direction of the slots prevents the yarn flying out. The wheels of Leonardo had two separate bands, one driving the spindle and flyer, the other driving the bobbin at different but constant speeds. In wheels of the present day (for

wheel-spinning is by no means extinct), driven in a similar manner, we have but one band and one wheel, going twice round the wheel, once round the wharve of the spindle, and once round that of the bobbin. Fig. 9 shows the type. France, Belgium, Holland, Germany, and Russia have wheels so constructed. The fact of having but one band causes equal strain and consequently maintains equal relative speed to both driven parts. In Fig. 9, and indeed in most types of wheels, there is bobbin lead, the bobbin revolving on the spindle. While it may be desirable to maintain equal relative speeds and not occasion any undue difference which might occur to either driven part by one band becoming slacker or tighter than the other, the system is not correct either in theory or practice. The relation in the speeds should be gradually and deliberately altered as the bobbin filling becomes larger in diameter. In the case of spindle-lead, the bobbin should be slowly checked; in bobbin-lead, the spindle should be checked.

As this Paper is not intended to deal with antiquarian lore, but rather with the principles of spinning, it is a matter of little importance which system, that of regularly maintained speeds or that of adjustable speeds, was first introduced in actual work; it would in any case be a question of speculation, but both modes can be distinctly traced back for over 300 years and have continued until to-day. Where an adjustable speed is employed and where there is spindle-lead the spindle alone is driven, the bobbin being carried round simply by the pull of the spun yarn; as the bobbin gradually fills and increases in diameter so its velocity increases in proportion, while the driven spindle runs at an ever constant speed. To bring both into proper relation, therefore, the most convenient way is to diminish the bobbin velocity. This is effected, Fig. 11, by means of a brake or drag, which is a short string *a* fastened at *b* or at any suitable place on the framework; the string is brought over the half circle of the groove in the bobbin and terminates on a peg *c*, round which it is wound and by which it may be tightened as a violin string is. The tighter the string is wound, the more it presses on the bobbin and occasions a brake or drag which is adjustable at the will of the operator. In the

case of bobbin-lead, where the bobbin is driven and the flyer and spindle carried round by the pull of the yarn, the drag is put on the flyer. In Fig. 10, *a* is the string passing over the flyer, *b* is the peg.

This latter system is mostly used where adjustable speeds are in operation; the author has only seen two types of wheels with flyer-lead. Fig. 11 belongs to a Bohemian wheel and the other was used in France. The majority of wheels, however, including those formerly used in these kingdoms, had both flyer and bobbin driven, and although incorrect they perhaps produced better results, taking all circumstances into consideration. If the relative speeds were so adjusted that correct working would take effect when the bobbin was half full, there would be very little variety in the yarn. When the bobbin is empty there would be too little tension and hence slack winding; when full the tension would be too great, causing an extra drawing and consequent greater attenuation of yarn; such would be the case in a modern spinning-frame. But a spinning-wheel has an adjustment which no modern frame attempts, namely, the adjustment of feed in relation to these matters; the spinstress has the flax under her absolute control and can guide to the flyer more or less as occasion demands. Where, however, a delicate drag is placed in the hands of one unacquainted with its working or unwilling to trouble with it, it may prove harmful and form itself an impediment to equally spun yarn.

These suppositions the author knows to be facts from personal observations. He has visited many cottages in different countries of Europe where wheels of this type were in work; and although he always inquired and often adjusted the wheels, he only met one spinstress who understood the meaning of this drag. As a rule the drag was loose and inoperative; sometimes it was broken off; in some places, as in Lombardy and Tuscany especially, they never attempted to put it on. A test recently made for the author at the Municipal Technical Institute, Belfast, on a dry spinning-frame, with a dragged bobbin and with a bobbin not dragged showed no variety in the twist, either at the beginning or

finish of the bobbin. With the drag a somewhat stronger yarn was produced, and certainly the building of one could not be compared with that of the other. The test, however interesting, is not comparable with a similar test on a spinning-wheel, for, although the drag weight was removed, there was still the friction of the bobbin base retarding its motion, which in the horizontal spindle of the wheel could not take place. The tension on the yarn was slight, and although it ballooned freely there were no breakages.

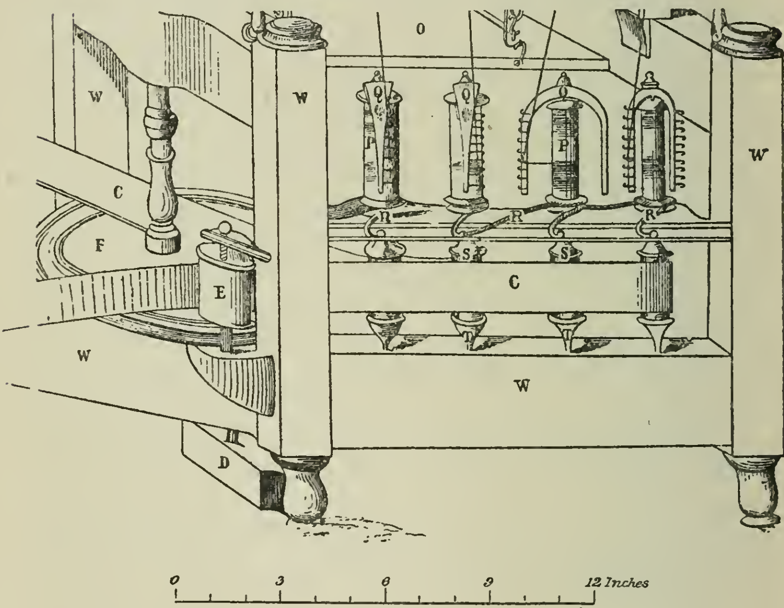
In a spinning-wheel without drag the yarn would incline to curl and have an appearance of overtwist. Here, too, the spinstress, if she is clever, can adjust the tension as she feeds the flax to the flyer. We have noted that in Leonardo's spinning-wheel he inserted pegs between the driving wheel and the band for the purpose of tightening the band. These pegs were removed or inserted at pleasure; if the band was tight the spindle ran quicker and naturally gave a greater twist to the yarn. In later spinning-wheels this is arrived at by means of what is called the temper-pin. Fig. 5, Plate 36, contains one example of this: *a* is a screw, called the temper-pin, passing through a hole in the upright *b* large enough to take the outside diameter of the screw easily; this upright has a slot a few inches long into which the framework *c*, which carries the spindle, bobbin, and flyer, is inserted. The screw or temper-pin engages with a nut on the part of the framework thus inserted, and the whole framework with its working parts can thus be raised or lowered, making the band slack or tight at pleasure; *d* is an adjustable support for the framework, which by means of a screw is raised or lowered to suit.

We now come to mechanical spinning. As already noted, all cotton yarn was spun on a wheel without flyer or bobbin, in its working parts similar to Fig. 6 (page 690). None of the yarn thus spun was strong enough for warps, hence linen yarn was always used for that purpose. The calicoes of a century and a quarter ago were in reality unions, the weft being cotton; in unions of to-day, however, the weft is linen. The mode of spinning was of necessity extremely slow and could not keep pace with the ever-increasing requirements of the looms; a more rapid mode of yarn production

therefore became necessary. As the author in this Paper does not deal with any part of a spinning-frame but the spindle and its attachments, he will refer only for a moment to the great problem that immediately presented itself, namely, how to produce in an inanimate machine the delicate and exquisite touch of the human fingers in drawing out with precise regularity the material to be

FIG. 12.

*Cotton-Spinning Frame.* (Richard Arkwright, 1769.)




spun. It looks simple now when one is accustomed to it, but many a brain was worried and many a scheme abandoned before the way was made clear. Once overcome, however, the details in mechanical spinning followed.

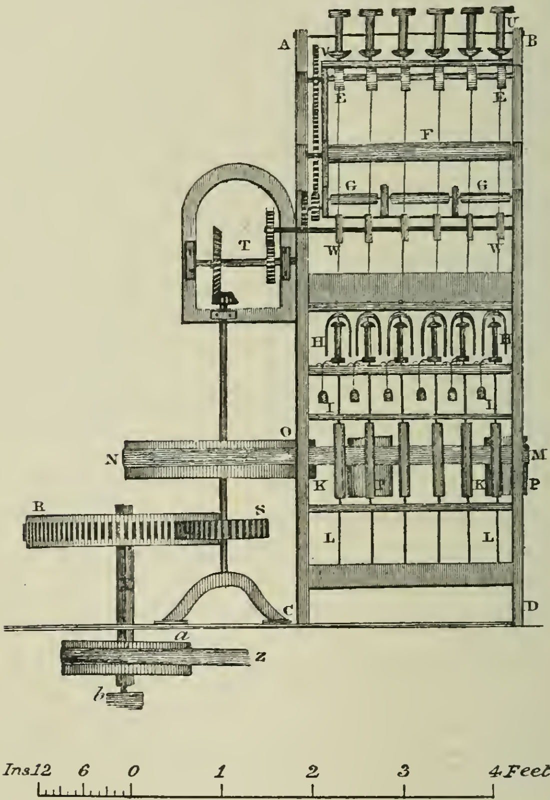
In 1769 Richard Arkwright brought out his cotton-spinning frame; the spindle he adopted was that of the spinning-wheel with flyer-lead and dragged bobbin set upright. Fig. 12

is from the original specification, and the following is the descriptive matter: "P the bobbins and spindles; Q flyers made of wood with small wires on the side which lead the thread to the bobbins; R small worsted bands put about the whirl of the bobbins to wind up the thread faster or slower; S the four whirls of the spindles; T the four spindles which run in iron plates." The material parts of Arkwright's invention were, of course, the principles of drawing. When this spinning-frame was set to work, yarns strong enough for warps were produced. This was due solely to spinning the cotton by means of a spindle and flyer. Hargreave's system, which embodied the principle of spinning with a bare spindle as in the cotton-spinning wheel, supplied, like that wheel, weft yarns. It seems a wonder, hardly capable of explanation, that the wheel without the flyer and bobbin was used up to the very last for spinning cotton yarns. Not only was the process slow, involving first the spinning, then a stoppage of that operation in order to wind, but the yarn turned out was only suitable for wefts. In Ireland and Scotland at that period there were thousands of wheels spinning linen yarns by means of the flyer and bobbin, such as Arkwright introduced into his frame. Even in Lancashire, then as now the seat of cotton spinning, there were many flax wheels at work. These were capable of spinning cotton quicker and better than by the older process. In Ireland spinners were very dexterous, wages were low and work often scarce, and yet no advantage was taken of these wheels. It would seem that custom had established the one mode for flax and the other for cotton, and, not till Arkwright's experiments proved it, was it known that the flax system was capable of spinning cotton.

Until 1775 yarns continued to be spun mechanically by this wheel system introduced by Arkwright into his frame. There was no building motion such as Leonardo da Vinci exhibited in his drawing; the spinning-frame had to be stopped at intervals in order to place the yarn in another hook and thus build up the bobbin; the system of dragging was of the violin peg type already mentioned. At some later intermediate period, cloth or leather washers were introduced upon which the vertical bobbin was placed;

as in worsted spinning at the present time, the greater the drag required the larger in diameter is the washer. The friction of the bobbin base against these materials retarded its velocity and thus constituted a more or less perfect drag.

FIG. 13.—*First Authentic Account of the Drag-weight.* (Described by Gray in 1819.)




The first intimation we have of the present-day drag-weight is indicated in Fig. 13,* when a band, at that time of leather, was

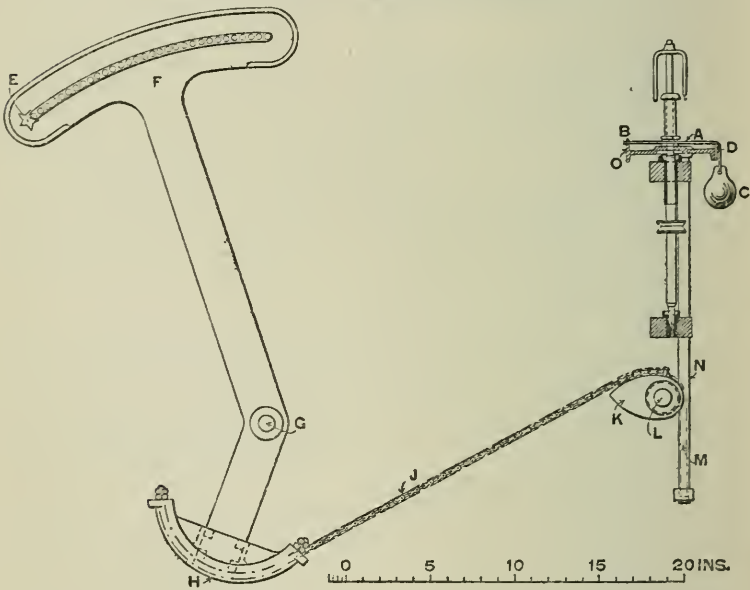
* Gray on "Spinning Machinery," 1819, Fig. 5, Plate VII.

fastened at the back of the spindle rail and brought round a groove in the bobbin base terminating at the other end in the drag-weight I. The adjustment of this weight to the left or right diminished or increased the drag by bringing the leather band with lesser or greater pressure against the groove in the bobbin base. Motion to the spindles was given by means of the belt M N which turned the rollers K K attached to the spindles; P P were iron rollers so placed as to keep the belt in touch with the rollers K K and adjustable, so as to tighten or slacken the belt, and thus permit the spindles to revolve quickly or slowly as was found necessary. Neither in this drawing nor in the descriptive matter attached to it is there any appearance of a building motion. The drag in a modern frame is constituted practically in the same manner as that in Fig. 13. This will be seen with reference to the last diagram, Fig. 14 (page 700). A is the drag band fastened at B, brought over a groove in the bobbin base, and having a drag-weight C attached to the other end; at D there is a serrated strip attached to the builder in the notches of which the drag bag is moved to give pressure more or less on the bobbin, and thus regulate its velocity.

It has been noted that in Arkwright's 1769 specification there was no building motion. In 1775 Arkwright again applied for and received a patent, No. 1,111, for various movements in connection with the treatment of fibres. In order to lessen the possibility of any rival secretly making use of his designs, he purposely drew up the specification in a manner so obscure that one can hardly penetrate it. The drawings attached are quite as unintelligible, being placed at random on the sheet and unconnected with neighbouring parts. As the Patent Laws require a clear and lucid interpretation so that one with a fair knowledge of the subject may understand it, Arkwright defeated his own ends. His patent was opposed, the Courts decided against him, and his ideas became the property of anyone. Attached to the specification there is a very involved drawing showing what purports to be a method of raising and lowering the bobbin on the spindle, evidently for the purpose of building the yarn equally. In the numerous specifications taken out after this date for the treatment of flax fibres in spinning, the

claims have reference primarily to the very difficult task of drawing that stubborn fibre to produce an equal yarn. Many details of the frames not distinctly relevant to that process are therefore absent in the specification drawings, the object being not to complicate the main idea with unnecessary detail. As in every instance the flyers are shown without hooks or other such means of regulating the

FIG. 14.
*Modern Spindle for Wet Spinning Frame.*




winding, and the spindle blade left long enough to accommodate the rise and fall of the bobbin, it may be assumed that some building attachment on the principle introduced by Arkwright was in use. From the above statement Fig. 13 is an exception. This is not, however, from a patent specification, but from a text-book on flax spinning published in 1819, the explanatory matter in which is given in a very detailed manner; it is possible that the rail on which the bobbins rest may have had an up and down movement.

In conclusion, the author again refers to Fig. 14 for a modern building motion, pinion E, revolving at a fixed point, engaging with teeth of segment F, and following the curve changes from the upper to the lower side of the quadrant teeth, and vice versa, producing a regular back and forward movement of the quadrant which is centred at G, and having on its base a segment H attached to a chain J actuating at its other end a cam K on a shaft L. Around a pulley on this shaft another chain M works, fastened to a small bracket on a round rod N fitted to a hole in the builder O, on which latter the bobbins rest, and which is thus traversed up and down to build the yarn evenly upon the bobbins.

The Paper is illustrated by Plates 35 to 38 and 4 Figs. in the letterpress.

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*Discussion.*

The President said the members had by their acclamation passed a vote of thanks to the author for his very interesting archæological Paper on a subject which was probably not very familiar to the majority of the members of the Institution. One of the most interesting facts brought to their notice was the connection of Leonardo da Vinci with the important development referred to. Leonardo was well known as a very wonderful man, who could do almost anything both in art and science; but the detail with which his sketch had been produced and the accuracy of his thought and design were really marvellous. He had often said that engineers must have imagination, and certainly Leonardo da Vinci had that faculty; his powers in that respect had rarely been equalled.

Mr. John Erskine, in opening the discussion, thought all the members were very much indebted to the author for his very interesting Paper. The author had quite distinguished himself in the interest he took in the ancient spinning wheel, and had taken up the subject very much from its archæological side. Personally, he was not able to claim much knowledge of what had been done in the past, as all his work had been connected with modern flax spinning. The President had said that Belfast possessed some of the biggest industries that were in existence; undoubtedly, one of the big things dealt with in Belfast was flax spinning. Although so much had been done in past ages in connection with flax spinning, the industry having come down to them from the old days in Egypt, there was still a very great deal to accomplish. The spinners in the modern mills had not been careless; they had tried to produce the best article out of the material available. The material was more troublesome to deal with than either cotton or wool, and consequently the manufacturers had had to contend with many difficulties in the endeavour to produce an article that would satisfy the merchant. He had been many times in offices on the Continent where the merchant buying goods from him would take a piece of linen, put it up against the window and point out all the little ticks in the material or even a thick thread. The whole of the imperfections of the material were pointed out, and when he had come out of the office he had thought, "How am I to please that man; the material must be so perfect?" It was his general experience that the men who wove the yarn had no idea of the way in which it was produced and the difficulties that were experienced, and consequently they just demanded perfection. To meet that demand for perfection, the spinners in Belfast had done a very great deal, as the author had stated. Like all other mechanical appliances at the present day, the spinners had been reaching forward a good deal; and there were men in Belfast, such as the author, who had a great love for their subject, and many of whom could mould machinery just as if it were wax in their hands for the purpose of producing that perfection of manufacture that was demanded.

The author had read a Paper on Flax Hackling * at the last Belfast Meeting of the Institution, to which he had given a great deal of anxious thought. Considerable ability had been shown by many people in their endeavour to deal with the subject, and as a result very perfect machinery had been produced, but nevertheless still further efforts were being made to produce a better article. With regard to cotton and wool, the spinners had been trying to make the most perfect yarn that could be produced, and they had demanded from the machine-maker the very best machinery for attaining that end. There were several machine-makers in Belfast who did their best in that way, and he thought they deserved more than credit for their efforts. It was a source of great pleasure to him that he had been accorded an opportunity of making a few remarks on the Paper, thanking the author for the labour he must have spent in its preparation, and saying at the same time how much he admired his industry and his love for the subject.

Mr. HENRY TAYLOR said the Paper opened up a very important question in connection with the development of the manufacture of small articles. That had played an important part in connection with the commerce and economics of the country as well as in its civilization, because he thought the people of the country would have been in an awkward state without the spindle. He was afraid that many people lost sight of the very important function that small articles played, notably on the textile industries, and particularly on wool, cotton, and linen.

Dr. H. S. HELE-SHAW (Member of Council) said he had known many inventors who were by no means capable of describing in intelligible language what they had invented. He was inclined to think that possibly Arkwright really did his best to describe the machine he had invented. Instead of going to some one whose business it was to make descriptions of inventions, he probably tried to do this himself, and the result was that his patent would not

* Proceedings 1896, Part 3, page 283.

(Dr. H. S. Hele-Shaw.)

stand. From all that was known of the history of Arkwright, he was apparently an honest man above everything else, and would be likely to do his very best to describe his patent properly if he could.

Mr. JOHN A. F. ASPINALL (Past-President) said that in reading the portion of the Paper dealing with Arkwright, he remembered pointing out at the Birmingham Meeting of the Institution held in 1910, that the first hank of cotton that was ever produced, except by human hands, was produced by John Wyatt, who was a workman in Boulton and Watts' factory at one time; but he afterwards left them and joined a man named Paul, who was his financial partner, and Wyatt and Paul sold their patents to Arkwright. Arkwright was not the originator of the invention but John Wyatt, and anybody who desired to see the first hank of cotton produced by mechanical means could see it in the Birmingham Museum, with the original memorandum attached to it, stating that it was the first ever produced.

Mr. JOHN HORNER, in reply, said he had very little to reply to, because the Paper was not a controversial one and did not lend itself to discussion. He was very much obliged to the various members who had spoken so favourably of it. The writing caused him very little trouble, as he had all the material collected for another purpose. In reference to Dr. Hele-Shaw's remarks (page 703), he thought there was no one who could say that sturdy Richard Arkwright ever did, or attempted to do, a dishonest action. Personally, he did not think that that concealment was at all dishonest. If Arkwright did at that time desire to conceal the various movements in connection with some very important patents, he did not think that was a dishonest action. Undoubtedly Specification No. 1,111 was in some peculiar way so constructed that very few people could understand the meaning of it, and when it was borne in mind that about thirteen years previous to that he had already taken out a patent for his rollers which were accurately described one could not, he thought, believe but that Richard

Arkwright had deliberately made his second specification obscure. He was perfectly justified in doing so if by those means he could keep the pirates away, because it was difficult in those days to trace piracy. The spinning machine was usually driven by a horse, and often driven by hand, and placed in various cottages—it might be in any obscure part of the country where the work could be carried out without Richard Arkwright knowing of it. On those grounds he thought Arkwright was perfectly justified in doing all he could to prevent his intentions being pirated or used by others.

The Paper was not written from an archæological standpoint, nor did it intend to treat with the subject from an archæological point of view; it was principally intended to treat of the elements which he thought were necessary to the understanding of any mechanical subject.

In reply to the very interesting point that Mr. Aspinall had raised (page 704), he said that Wyatt and Paul undoubtedly introduced the system of drawing by roller many years previous to Richard Arkwright doing so. In their first specification they described a very extraordinary mode of drawing with one pair of rollers retaining while another pair of rollers were drawing and rotating (to give a slight twist to the cotton) at the same time—a fantastic and utterly impossible mode of working. In their second specification they introduced solely a retaining roller; there was no drawing roller whatever, the draft being produced by the spindle. Arkwright's principle consisted in the employment of retaining and drawing rollers, such as were used at the present time. Anyone who looked at Wyatt and Paul's second specification, which was the only practical one they ever worked by the aid of horse traction—he thought at Birmingham—would see they had a roller which passed the cotton down, and then it was drawn by the spindle and flyer. Arkwright introduced a means of retaining rollers, that is, he had a couple of rollers which retained or held the rovings, and those were revolved at a certain speed. The drawing rollers which drew out the fibres were revolving at six or seven or eight times the speed, thus attenuating or reducing the roving so much, and spinning it then directly to the bobbin. To Arkwright must

(Mr. John Horner.)

therefore be given the credit for being the man who introduced a really practical system of retaining and drawing rolls; and to Arkwright must also be given the credit for being the first man who ever made a practical success of cotton spinning. It was thirteen or fourteen years prior to Arkwright's first patent that Wyatt and Paul brought out their system, but they made absolutely no success of it.

# WIRE ROPES FOR LIFTING APPLIANCES, AND SOME CONDITIONS THAT AFFECT THEIR DURABILITY.

By DANIEL ADAMSON, *Member*, of Hyde.

The question of the durability of the parts of mechanical structures seems to be strangely neglected by all authorities. A designer has generally the choice of several formulæ for calculating the mere strength of a given member, but usually he has to depend upon his own experience for the correctness of the proportions that will ensure for it a reasonable length of life. The durability of wire ropes in particular is of great importance to all engineers, whether engaged in the design and manufacture of lifting appliances, or in their care and management.

The two most important conditions appertaining to the manufacture and use of steel wire-ropes that affect their durability are:—

(*a*) Quality of material and size of wire.

(*b*) Diameter of pulleys and arrangement of ropes.

(*a*) *Quality of Material and Size of Wire.*—The wire used for lifting ropes is of steel having an ultimate tensile strength varying from 80 to 130 tons per square inch. Although ropes made from material having a high tensile strength are of smaller diameter, for a given load and a given factor of safety, yet this is not a great advantage to the crane designer because the stiffer character

of the wires makes larger drums desirable, if the durability of the rope is to be considered, notwithstanding that some rope-makers claim as an advantage for the stronger material that it does enable smaller pulleys to be used with a consequent lower cost of the working parts of the crane.

The ratio of the diameter of the individual wires to the diameter of the completed rope is an important factor. If the wires are too large they are stressed considerably when passing over the pulleys, and accordingly the material is quickly fatigued and the wires break. Smaller wires, on the other hand, are more quickly worn through by rubbing against the pulleys and against their neighbours in the body of the rope. The stress in a wire due to bending round a pulley is directly proportional to the modulus of elasticity and to the diameter of the wire, and inversely proportional to the radius of the pulley; therefore the radius of the pulley should be increased with an increase in the modulus of elasticity, if the same number of bends is to be endured by a stronger wire of the same diameter. Unfortunately a theoretical calculation of the stresses induced in the wires of a rope by being bent over a pulley does not alone afford a reliable guide to the length of life to be expected from the rope, for consideration must also be given to the mutual wear that takes place amongst the wires.

Assuming for the purpose of comparison that two ropes are constructed of equal size, one from wires half the diameter of those in the other, then for equal strength the one rope will have four times the number of wires and each of the wires will have one-quarter the cross sectional area. According to the usual formula, the stress due to bending will be half as severe in the smaller as in the larger wires, when the ropes are bent over pulleys of the same diameter. If it be allowed that a reasonable figure for the estimated stress due to bending an ordinary rope over a pulley of a size usually adopted in crane design be, say, 30 tons per square inch, and the stress due to the suspended load be 10 tons per square inch, there will be a range of stress of 40 tons per square inch in the material each time the maximum load is lifted and released, and the corresponding stresses in the rope of finer

wires will be 15 tons per square inch due to bending, and as before 10 tons per square inch due to the suspended load, or a total range of 25 tons per square inch.

Judging by the discussion that took place on Messrs. Eden, Rose and Cunningham's Paper before this Institution in November last on "The Endurance of Metals," there is, as yet, no agreement as to the exact effect upon the endurance of variations in the working stresses. It seems, however, to be reasonable to assume that a reduction in range of stress from 40 tons per square inch to 25 tons per square inch would increase the life of material, such as ropes are composed of, about 500 times. As no such improvement in the life of a rope has ever been experienced, or is to be reasonably expected, it must be taken for granted that abrasion is the principal factor in limiting the life of wire ropes, and therefore the effect of abrasion upon the suggested rope of finer wires may now be considered.

When the rope of finer wires is passing over the pulley, there being four times as many wires in it, the pressure at each point of contact between the rope and the pulley and between the individual wires of the rope may be assumed to be one-quarter of what it is in the rope of larger wires. The wires being of half the diameter the damage done to them by contact, even under this lower pressure, will be at least half as much as occurs to the coarser wires in the other rope, and this half damage done to a wire of one-quarter the sectional area will result in the cutting through of the wire in half the time, so that the effect of abrasion upon the rope of finer wires will be twice as great. If a smaller pulley be used for the rope of finer wires, as suggested by some authorities, the pressure at the points of contact and the stress due to bending will be proportionately increased, so that it may reasonably be expected that, with a pulley-diameter bearing the same proportion to the diameter of the wires, the life of the rope with fine wires will be one-quarter of that of the rope of coarser wires working over a pulley of correspondingly increased diameter.

A German investigator (Ernst Heckel) refers to the very great surface pressures on the wires at the place of contact with

the pulley (amounting in his opinion to as much as 12 tons per square inch) as a vital point in connection with the wear of wire ropes. This high pressure, accompanied as must be the case by relative movement even if quite small, readily accounts for the wear which takes place on the surface of the wires where they touch the pulleys or the other wires in the rope.

(*b*) *Diameter of Pulleys and Arrangement of Ropes.*—The lists issued by makers of wire ropes contain recommendations as to minimum sizes to be adopted, but no information is given as to the effect of using pulleys of different diameters. The author has felt for many years past the want of such information: the experience of users afforded no reliable guidance, presumably on account of the great difference in the conditions under which ropes work in different shops. Reference to a Paper read before the Manchester Association of Engineers by Mr. Matthews in 1902 brings to light one great difference in the working of cranes. Mr. Matthews, in his Paper, suggested that 400 to 1,700 lifts per crane per annum was the amount of duty required from certain cranes under his control, while the present author, in the discussion on Mr. Matthews' Paper, mentioned 32,400 to 43,200 lifts per crane per annum as representing his own experience in another class of work. Other important features that will affect the life of a crane rope are the average weight lifted and the average height of lift; cranes are generally occupied with loads much below their nominal capacity, but this will vary in different workshops as will the proportion between the maximum height of lift available and the height most frequently attained by the hook.

Inquiries addressed to the users of cranes elicited very various replies; ropes working upon cranes of the same general design were found to last for periods of from two years to ten years and upwards, and one correspondent suggested that 20 years might be expected from ropes on cranes (of from 5 to 20 tons capacity) if damage from accidental causes could be eliminated. As might be expected, the ropes on foundry cranes have not so long a life as in erecting shops, the relative difference being perhaps as three is to five.

The most reliable and consistent information that the author has been able to discover (with the assistance of numerous friends and correspondents, and also of the library staffs of the Institution of Mechanical Engineers in London and of the Engineering Library in New York, to all of whom his sincere thanks are due) is contained in a Paper by Mr. A. S. Biggart * published in 1890. The experiments to which this Paper refers were undertaken with the object of selecting the best form of rope to be employed in the construction of the Forth Bridge. A full description of the apparatus used and the details of the investigation will be found in the original Paper, and the present author will content himself with a short reference to the experiments and an abstract from the conclusions arrived at, adding some deductions he has made for his own guidance and for the purpose of this Paper. The apparatus used by Mr. Biggart contained two pulleys, round which the rope under trial was passed, the lower pulley being weighted to give the required tension on the rope. The experiments consisted in passing the ropes, under a normal working load, to and fro over the pulleys until breakage ensued. Experiments were repeated with different diameters of pulleys and different makes of rope, and the accompanying diagram, Fig. 1 (page 712), shows the life of different classes of rope as affected by the diameter of the pulleys.

The effect of oiling the ropes is shown by the diagram to be very beneficial, increasing the life of a given rope by two or three times. This is obviously due to the reduction of the cutting action of the wires upon each other. Experiments were also made to ascertain the effect on the life of a rope of running it over pulleys so arranged that the rope was subjected to reverse stresses, Fig. 4 (page 716). The results obtained from this series of experiments showed that generally the life of a rope working under such conditions was only one-half as long as a similar rope bent in one direction only.

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* Proceedings, Inst. C.E., 1890, vol. ci, page 231.

FIG. 1.

*Experiments on Durability of Wire Ropes as affected by Diameter of Pulley* (1890).

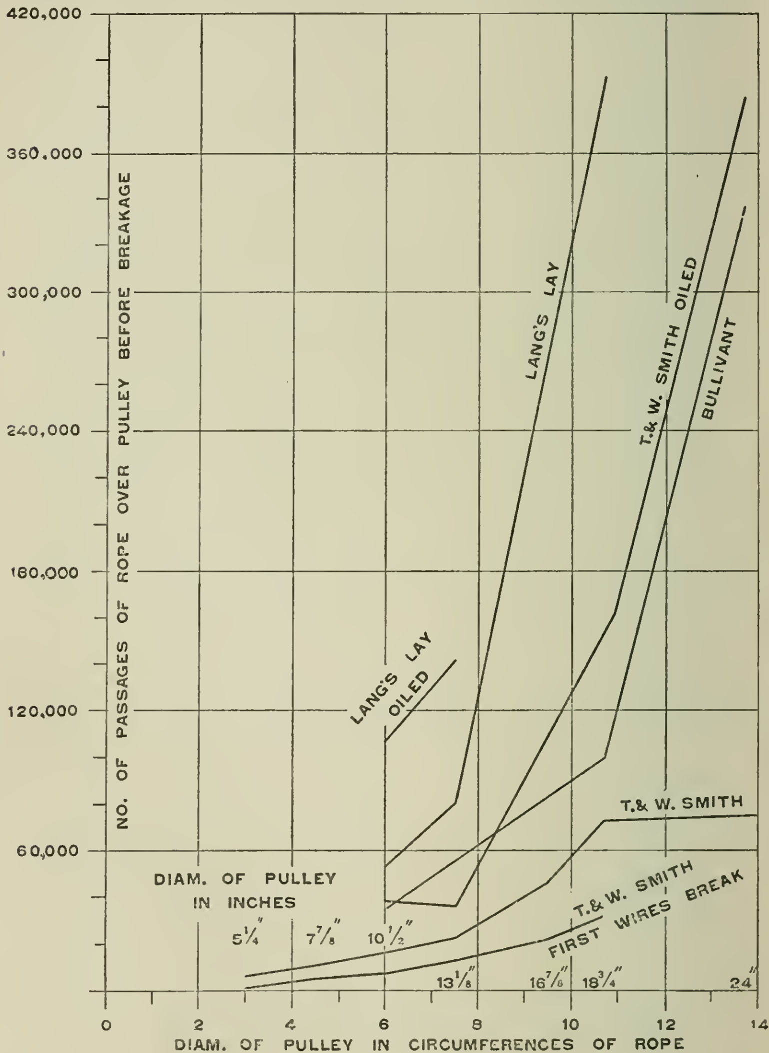


Fig. 2.—*Regular Curves based on data in Fig.* 1.

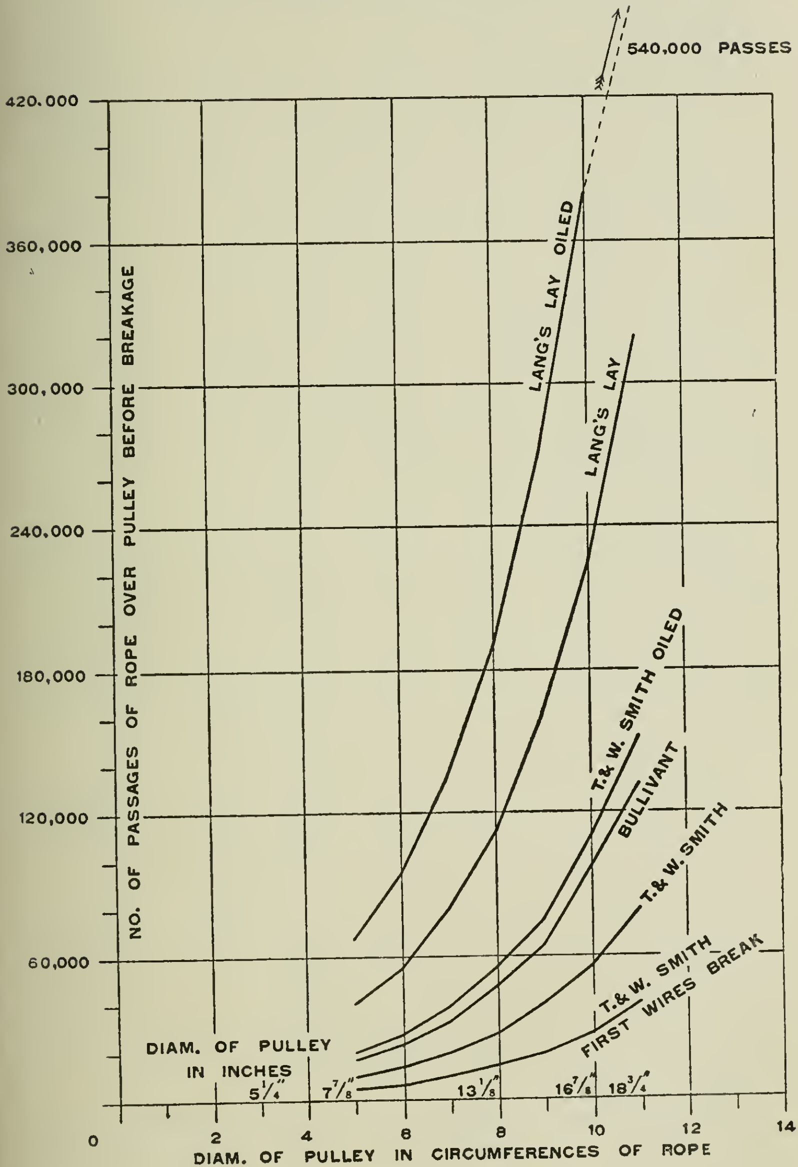
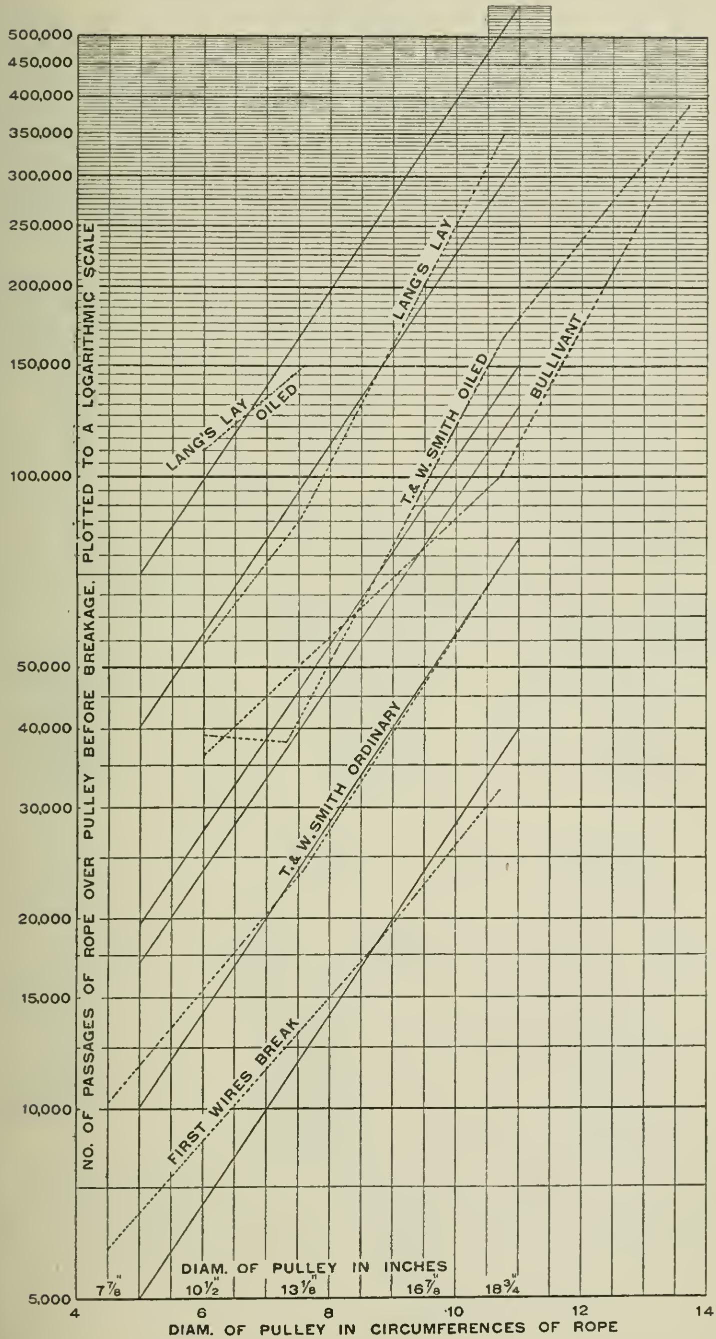


Fig. 1 is based upon the actual figures tabulated in Mr. Biggart's Paper, while Fig. 2 (page 713) shows the present author's approximations, as obtained by the simple method of drawing fair and regular curves through or near the points representing the results of Mr. Biggart's experiments over such a range of pulley diameters (measured in terms of the circumference of the ropes) as obtain in general overhead crane practice. Several interesting deductions may be drawn from a study of these figures. The time of breakage of the first wires of a rope in the lowest curve is only recorded for one make of rope, but comparing it with the second curve, which shows the time of breakage of whole ropes of the same make, it will be seen that when the first wire breaks the rope may be assumed to have passed through one-half of its life, and as no one knowingly works a rope until it breaks entirely, then the breakage of even a few wires is a sign that a rope should be carefully watched and replaced by a new one at an early opportunity.

The effect of varying the proportions of diameter of pulley to diameter of rope is one of the most important features to be noticed. Speaking generally, Mr. Biggart's experiments show that increasing the diameter of the pulleys by an amount equal to two circumferences of the rope will double the life of the rope. This is approximately correct for all the varieties of rope and conditions experimented with, and may therefore be taken as equally correct for all the varying conditions under which cranes are worked. It is very remarkable that so simple a rule should evolve from such numerous and varied experiments, and the author hopes that its statement in this form will be of some value to designers and other interested members. That it is sufficiently correct for all practical purposes may be readily seen by referring to Fig. 3 (page 715), where the ratios of pulley diameters to ropes are plotted as abscissæ to a linear scale, while the durability of the ropes is represented by ordinates drawn to a logarithmic scale.

These conclusions enable one to express a definite value for the effect upon the durability of ropes, of the various arrangements

FIG. 3.—*Durability as Affected by Diameters of Rope and Pulley.*



Various Arrangements in Lifting Appliances.

(*See* Table 1.)

FIG. 4.

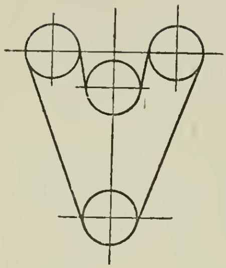


FIG. 5.
*One Bend.*

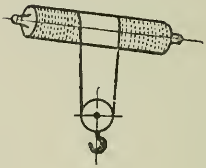


FIG. 6.
*Three Bends.*

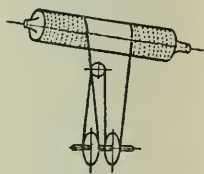


FIG. 7.
*Three Bends.*
*One Reverse.*

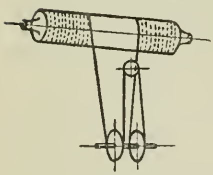


FIG. 8.
*Seven Bends.*

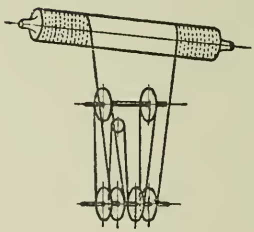


FIG. 9.
*Eleven Bends.*

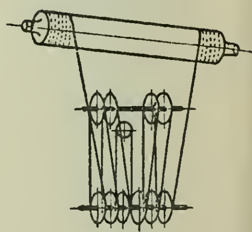


FIG. 10.
*Seven Bends.*
*One Reverse.*

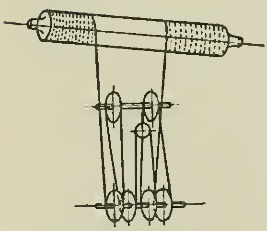


FIG. 11.
*Eleven Bends.*
*One Reverse.*

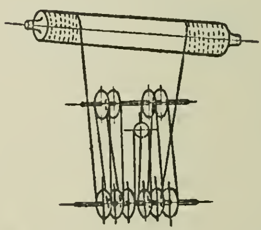
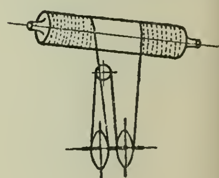


FIG. 12.
*Three Bends.*
*Large Bottom Pulleys.*



of pulleys that are commonly adopted in overhead cranes, some of which are illustrated in Figs. 5 to 11. Assuming that Fig. 6 in which the ropes make three bends in working, namely, one at

the upper drum and one on each side of the lower pulley, i.e., at entering and leaving) is the arrangement most frequently adopted in practice, and representing the anticipated life of the rope under these conditions by 100, then the relative lives of the ropes in each of the other arrangements indicated will be shown in Table 1.

TABLE 1.

*Comparison of Anticipated Length of Life of Ropes arranged as shown in Figs. 5 to 11.*

| Fig. No. | Number of Bends. | Relative Life of Rope. |
|---|---|---|
| 5 | 1 | 300 |
| 6 | 3 | 100 |
| 7 | 3* | 75 |
| 8 | 7 | 43 |
| 9 | 11 | 27 |
| 10 | 7* | 37½ |
| 11 | 11* | 25 |

* Including one reverse bend which is twice as effective in wearing out the rope.

If it be desired to design each of the above arrangements of pulleys so that the ropes shall have equal durability, then the ratio of the drum diameters to rope circumference (if the law indicated by Figs. 2 and 3 is to be relied upon) must be increased as shown in Table 2 (page 718).

It is quite usual for purchasers to specify in their inquiries that the diameters of the pulleys and drums must bear a certain relation to the diameter of the rope, but the author wishes now to emphasize the point that this stipulation is not sufficient in itself,

## TABLE 2.

*Required increase in Diameters of Rope Drums (measured in terms of Circumference of Rope) required to give Equal Durability.*

| Fig. No. | Increase over Diameter called for by Fig. 6. |
|---|---|
| 7 | 1 Circumference of Rope. |
| 8 | $2\frac{1}{2}$ Circumferences of Rope. |
| 9 | 4 ,, ,, |
| 10 | 3 ,, ,, |
| 11 | 4 ,, ,, |

without some consideration being also given to the arrangement of the rope and pulleys.

If the generally accepted ratio of seven circumferences, or twenty-two diameters, of the rope for the diameter of the barrel be assumed as suitable for the drum and pulleys arranged as in Fig. 6, then the diameters for the other figures, to give equal durability, should be as shown in Table 3.

## TABLE 3.

*Ratio of Diameter of Pulleys and Drums to Circumference of Rope to give Equal Durability.*

| Fig. No. | Ratio of Pulley and Drum Diameter to Rope Circumference. |
|---|---|
| 5 | 4 to 1 |
| 6 | **7 to 1** |
| 7 | 8 to 1 |
| 8 | 9·5 to 1 |
| 9 | 11 to 1 |
| 10 | 10 to 1 |
| 11 | 11 to 1 |

To make the comparisons quite fair between the different arrangements, it must now be pointed out that, owing to the increased number of falls of rope adopted in Figs. 8 and 9, the size of the rope may be reduced as shown in Table 4 while retaining the same factor of safety.

## TABLE 4.

*Relative Rope Circumference allowing for Smaller Ropes due to increased number of Falls.*

| Fig. No. | Number of Falls. | Relative Rope Circumference. |
|---|---|---|
| 5 | 2 | 140 |
| **6** | **4** | **100** |
| 7 | 4 | 100 |
| 8 | 8 | 70 |
| 9 | 12 | 57 |
| 10 | 8 | 70 |
| 11 | 12 | 57 |

Combining the figures given in Tables 3 and 4 will give drum and pulley diameters as shown in Table 5.

## TABLE 5.

*Drum and Pulley Diameters resulting from a combination of Tables* 3 *and* 4, *and still assuming that* 100 *represents the Condition in Fig.* 6.

| Fig. No. | Ratio of Pulley and Drum Diameter to Rope Circumference according to Table 3. | Relative Circumference of Rope as per Table 4. | Resultant Pulley and Drum Diameter assuming Fig. 6 = 100. |
|---|---|---|---|
| 5 | 4 | 140 | 80 |
| **6** | **7** | **100** | **100** |
| 7 | 8 | 100 | 114 |
| 8 | $9\frac{1}{2}$ | 70 | 95 |
| 9 | 11 | 57 | 90 |
| 10 | 10 | 70 | 100 |
| 11 | 11 | 57 | 90 |

The noticeable feature in the last Table is that whether two, four, or six falls are adopted, the diameter of the drum and pulleys should remain about the same, if the ropes are to have equal durability (compare Figs. 8 and 9 with Fig. 6). A recent text-book upon the subject of crane design states (as an advantage of a large number of falls of rope) that the proportionately larger

pulleys and barrel will ensure long life for the ropes, but the author hopes that he has made it clear that very large proportions are necessary to ensure a reasonable life for ropes on cranes with many falls of rope. Reference to Fig. No. 7 and Fig. No. 10 in Table 5 shows the increase that should be made in the diameter of the drum and pulleys if a reverse bend occurs in the run of the rope.

Another important detail in crane design may now be referred to. In Fig. 6, as already mentioned, the ropes make two bends at the lower pulleys to one at the drum, and therefore, if the lower pulleys are made of the same diameter as the drum, they will be responsible for two-thirds of the wear and tear of the rope. Now it is usually difficult to increase the diameter of the working barrel or drum of a crane, because to do so affects the ratio of the gearing and also requires a much larger framework with a correspondingly greatly increased cost of manufacture, but if it is agreed, as a result of Mr. Biggart's experiments, that increasing the diameter of the pulley, over which a loaded rope passes, by an amount equal to twice the circumference of the rope, reduces the evil effects of bending the rope round it to one-half, then a simple means of improving the durability of crane ropes is immediately at the disposal of the designer, namely, to increase the diameter of the pulleys in the blocks, leaving the drums of the original size, as indicated by Fig. 12 (page 716). This alteration can usually be effected without serious alteration of the design, and may even be carried out on existing cranes.

The result of increasing the diameter of the pulleys (as shown by Fig. 12) by an amount equal to two circumferences of the rope, will be that the effect of the double bend round the lower pulley is halved, and the resultant effect of the three bends will be equal to two only and the relative life of the rope will be increased by 50 per cent., or the drum diameter might be reduced by an amount equal to 1·2 times the circumference of the rope with a corresponding reduction in the size of the framework of the crab or winch, while still retaining a relative life for the rope equal to Fig. 6. In this case the diameter of the lower pulleys would only require to be about one circumference of the rope larger than the original size of Fig. 6.

In making the foregoing comparisons of diameters of drum and pulleys with different arrangements of rope, it has been assumed that the hook is raised to the full height available at each lift. This however is not the case in actual practice, the majority of loads not being raised one-half this height.

This consideration brings to light another great advantage of Fig. 12 as compared with any of the others. Where, as is usually the case, the average height of lift in a shop does not reach half the maximum available, then that portion of the rope which passes under the lower pulley does not reach the upper drum, and accordingly is only subject to the wearing action of the two bends at the lower pulley. If therefore the effect of the bends at the lower pulley is reduced to one-half, by the proposed increase in diameter of the pulley, then the actual life of the rope will be doubled, instead of only being increased by 50 per cent. as was first assumed.

Where there are more than two falls of rope, as in Figs. 8 and 9, the effect of increasing the diameter of the pulleys by an amount equal to two circumferences of the rope is also very marked, reducing the effect of the seven bends in Fig. 8 to four and a half, with corresponding increase in the lift of the ropes. This shows up the fault of those designers who adopt large drums (in order to obtain the great length of rope entailed by high lifts) and are yet content to make the pulleys of small sizes, when they could enormously increase the durability of the rope by the adoption of larger pulleys at little extra cost.

When the rope makes a reverse bend at the barrel as in Figs. 7, 10, and 11, the barrel ought to be increased in diameter to counteract the effect of the reverse bend. Thus, if in each of these cases the diameter of the drums were made larger by an amount equal to two circumferences of the rope, the durability of the rope would be equal to Figs. 6, 8, and 10 respectively.

Some Continental makers point out, very rightly, the desirability of making the compensating pulleys of reasonable size. The motion over such pulleys is apparently considered as negligible by some designers (judging by the forms of construction adopted), but

this point of view overlooks the movement of the rope due to the swinging of the load, and the repeated bending of the rope at the same place over a small radius has an appreciable effect upon the durability of the rope.

Although the deductions laid down here appear too simple to need elaboration, a glance at the designs of many modern cranes shows that neither the designers, nor the purchasers, are aware of the importance of the principles involved, otherwise we should not see modern cranes in this country with reverse bends in the ropes, and as many as eight plies of rope to carry the load on cranes of only 15 tons capacity, while at the recent Brussels Exhibition there were cranes exhibited by well-known Continental makers showing the same faults.

The author would like to add that while he is aware of many conditions affecting the durability of ropes other than those he has referred to, he regrets that want of first-hand experience prevents him from dealing with them as he would like, and he hopes that other members will help to make up the deficiency.

The qualities of wire used vary considerably, and this, together with the heat treatment in manufacture and the care taken by the makers in testing and examination, are questions that makers of ropes are in a better position to discuss than users.

The "lay" of the strands and the lubrication of the rope when in use have each a considerable effect upon durability, and some guidance on these points may be obtained from Fig. 3 (page 715), where "Lang's lay" is shown to have more than double the life of ropes of ordinary "lay," and ropes that are oiled last more than twice as long as when this precaution is neglected, as already mentioned on page 711. The superiority shown by "Lang's lay" naturally gives rise to the question as to why it is not exclusively used, and the answer the author has obtained from rope-makers is that such ropes must be very carefully handled to avoid "kinks," and also they are found to be more liable to "spin."

The Paper is illustrated by 12 Figs. in the letterpress.

*Discussion.*

The President, in moving a hearty vote of thanks to the author for his very excellent and valuable Paper, said it dealt with a subject which required a great deal more research on scientific lines than had been given to it in the past. Every one who had to do with the use of wire ropes for lifting appliances knew the great uncertainty attaching to their life. Personally, he felt that after an engineer had done all he could in the way of designing the pulleys, and carefully constructing ropes from proper material, it was necessary to rely eventually upon careful inspection during use. Every law that had so far been laid down, even including that of the author's, would be very uncertain in practice, though, of course, of very valuable assistance to the designer. He hoped the Meeting would, in the course of the discussion, hear some experiences of the different systems of using ropes and their manufacture.

The resolution of thanks was carried with acclamation.

Mr. Robert Matthews (Member of Council), in opening the discussion, said he desired first of all to render to his old friend and neighbour, Mr. Adamson, his hearty thanks for taking the trouble to write the Paper. The author had gathered together some valuable information, but he was afraid the conclusions arrived at did not convince him that they were correct. He alluded more particularly to the illustrations on page 716 and Table 1 (page 717). The members would notice that in Fig. 5 there was a drum with two bends on it, and the pulley down below, and the author described that as one bend. Then Fig. 6, with the same drum arrangement, and the ropes arranged with the addition of two pulleys, the author called three bends. That, to his mind, did not seem to be right, as with a compensating pulley of ample size there was very little wear and the system then became only a two-bend, and consequently, as those two Figures were the basis for

(Mr. Robert Matthews.)
what the author determined later on, he was afraid the results would be somewhat fallacious and misleading. In one part of the Paper the author said that the compensating pulley wore, and he took that into consideration to the same extent as one of the pulleys. If he did it there, he ought to do it also in Fig. 5, but he did not. The author called that one bend, and that was where he thought he was a little unreasonable. That being so, he felt compelled to disagree with the foundations of the whole of his deductions later on in the Paper. He thought most of the members recognized that Lang's wire ropes were the best crane ropes. Personally, he knew Mr. Lang very well and he had used Lang's wire ropes very largely throughout his works, and had always found them very satisfactory indeed.

There was one point dealt with in the Paper about which he felt a little bit hurt. The author had probably written what he had done in an unguarded moment. The statement was made (page 710) that he (Mr. Matthews) suggested, when he read his Paper in 1902 before the Manchester Association of Engineers, that 400 to 1,700 lifts per crane per annum was the amount of duty required from certain cranes under his control, while Mr. Adamson in the discussion on the Paper mentioned 32,400 to 43,200 lifts per crane per annum as representing his own experience in another class of work. It would be preposterous for an engineer to suggest that. What he did was to give the actual number of lifts in a large gun shop where very large tools were used, and where, in one lathe, there would be perhaps three lifts in three weeks. If an expensive electric-driven crane was standing idle for a considerable amount of its time, it was not always economical engineering to go in for such an expensive piece of machinery, and he was only showing from actual experience that he had done the wrong thing by putting in an electric crane. If he had kept to the old type of rope crane, it would have cost less money and would have been more economical. It was not because he recommended that type of crane; it was merely a statement of fact. In some of the other shops there were electric cranes working as quickly as ever they could both day and night. In the iron foundry they changed their old rope cranes to

electrical cranes, carrying out the work themselves. Good large pulleys were installed, and they had been working now for over seven years with continuous overtime without a single failure in the ropes. His firm generally carried out the practice that was recommended very largely by Newall. He thought it was better to use a larger size in the way of pulleys, and his own practice was to use from 20 to 25 times the size of the rope. If that proportion were used, fairly good working and a good life for the rope would be obtained. With the 50-ton cranes in the foundry to which he had already referred, very heavy castings were made. The cranes were low, because it was mostly bed work that was carried out. Consequently the crane was very near to the heat, and also to the sand and dust. Notwithstanding that, the cranes had been working for over seven years, very often with a load of 70 tons on a 50-ton crane when very large castings were being made.

One other remark made by the author which seemed to him a little illogical was, that a double bend in a rope shortened its life by one half. Personally, he thought it would be less than that, because bending in both directions was more injurious than bending in one direction and pulling out straight. He believed that if actual figures were taken, it would be found that it was less than a half. He was sorry he had not been able to study the Paper more closely, but if the Paper and its criticism helped the members in coming to a right conclusion in their use of wire ropes, he would be more than satisfied.

Mr. A. Basil Wilson said that in listening to the Paper the point that struck him most forcibly was the comparatively small importance that the author attached to the question of reverse bends in ropes. Reverse bends did not, as a rule, occur in cranes used for foundry and other purposes where the direct load was immediately below the source of power, but wire ropes were used very extensively in connection with passenger and other lifts, in which it was quite a common practice for makers to pass their ropes round one if not two reverse bends. Such an example existed in the front passenger hoist of the building in which the

(Mr. A. Basil Wilson.)

Meeting was being held, and afforded an opportunity of estimating the effect on the rope. As originally put up, the hoist contained two reverse as well as three primary bends. The result was that the ropes lasted approximately not more than one session of about nine months, making 400 or 500 lifts a week. It was subsequently altered, and the lifting apparatus put at the top of the house, so that there was only one bend in the ropes and that a direct one. Since this was done no deterioration had occurred. The ropes had remained externally as they were, with practically no wear unless in the centre of the rope due to friction of the parts.

He would very much like to have the author's opinion, and the opinion also of the members, on the deterioration that might be expected to arise from a reverse half-bend. It might be assumed that it was 50 per cent. of that of a complete reverse bend, but he agreed with the last speaker that the effect of the half reverse-bend was much greater than a reduction of the life of the rope by 50 per cent. It was probably more in the neighbourhood of 70 per cent., especially if the reverse bend was one over which the rope passed for its entire length. If it was merely a reverse bend which was used for a compensating pulley, in which the travel of the rope round the pulley was small, then it might be almost neglected. Some years ago he had occasion to erect an engine of about 200 h.p. where it was desired to have brake efficiency records, and where a well-known system was adopted for the purpose in view. The ropes passed round two reverse pulleys, the intention being to ascertain the brake load on the engine continuously by the use of a recorder indicating the difference in the position of the reverse pulleys as they varied with the load. The apparatus worked well so far as ascertaining the brake horse-power was concerned, but was unsatisfactory with regard to the ropes. These ran at 4,000 feet per minute (there were eight of them), and began to show signs of deterioration in three weeks. Obviously that system had to be abandoned, and the ropes were taken direct, under which conditions they lasted about eight years, when the arrangement was changed and the rope drive abandoned.

The President inquired whether the ropes were of cotton.

Mr. Basil Wilson replied that they were steel ropes of $\frac{3}{8}$-inch diameter. But the special point he wished to emphasize was how far in the opinion of the members half-reverse bends influenced the life of a rope, which in many cases it was impossible to avoid.

Mr. J. Hartley Wicksteed asked whether Mr. Basil Wilson meant by a half-reverse bend 90° instead of 180°.

Mr. Basil Wilson replied in the affirmative.

Dr. William H. Maw understood that Mr. Wilson meant in another plane.

Mr. Basil Wilson replied that that was the case. Theoretically it might be assumed that the degree of the reversal would be the measure of the amount of wear in the rope, but it was a matter only experience could decide. He was sure the members of the Institution appreciated that the Paper was one of the most valuable character, since it dealt with a question which entered very largely into the everyday practice of the engineer.

Mr. J. Hartley Wicksteed (Past-President) said the President had made the very significant remark that the safety of ropes could only be ensured by proper inspection. He himself thought there was everything in that remark, because people did not quite realize the fact that if one single wire was broken in a rope, it upset the whole structure and was the beginning of the end. If all the constituent wires had been bound together to pull parallel with each other in straight lines, then the breaking of one wire would only reduce the strength of the rope by an aliquot proportion; but in fact the wires were spirally wound into strands, each complete strand was built up round a core, and each constituent wire was restrained by all the other wires in that strand from altering its pitch. If a single wire broke in this

(Mr. J. Hartley Wicksteed.)
one strand, the original structure of that strand was impaired and as soon as the loose wire worked out of its position, the curves of the other spiral wires would tend to straighten and the whole of that strand to lengthen, and to leave off doing its share of the work with the other strands. And when that strand had lengthened and worked out of its original position, it, in its turn, left off restraining the other strands, and the whole rope must lose its symmetry and become deformed. In other words, when a single wire broke, the rapid destruction of the whole rope was assured, if it kept on working under the same conditions as had broken the first wire, when the structure of the rope was perfect as a whole. He had tested a few ropes, one of which had 364 wires. It was a rope of about 8 inches circumference. By very careful attachments, every wire was in an equal state of stress, the rope broke clear of the attachments with 280 tons load and with a sound like the report of a heavy pistol. Of course the element of wear did not come in, in that direct test; but in the direct testing of ropes which he had made, he had always found that the gripping of every single wire, without any disturbance of the structure by splicing or otherwise, was essential for ascertaining the veritable strength of any rope, whether of wire or of hemp. He had tested hemp ropes spliced with eyelets for attachment, and out of a large number of tests, very few gave more than 5 tons ultimate strength. By means of certain attachments, which did not involve splicing or disturbing the coil of the rope, and did not involve the cutting or wounding of a single fibre in the rope, the loads went up to the region of 7 tons. He did not think the same care had been taken in the attachment of ropes in actual practice as had been found necessary for the attachment of ropes in order to prove them to their full capacity.

No doubt in mines or cranes the bending of the rope would destroy it before the attachments were destroyed. In deep mines, however, such as there were in South Africa, which were getting on for nearly a mile deep, some important wear of the rope took place near the attachment, on account of a tendency in the long rope to twist one way and the other way as it lengthened and shortened.

He did not know whether he ought to go any further into the subject of testing, but he might just mention that a machine had been devised, and experimented with on a small scale, which would put, in ten minutes, as much wear upon a rope as it might take ten years of actual work to do. It seemed to him that a great deal of time was wasted in making investigations and in collating the experience resulting from accidents and failures, which might be profitably saved by the spending of a few hundred pounds on an efficient machine, by which experiments could be made in a short time and without accidents.

He had found in his experience that machines could make discoveries. He would not say that they could make inventions, but they certainly could make discoveries. A machine could make discoveries that clever men had spent years over in searching for without discovering. A machine that was so constructed as to make a mechanical record of everything that happened—a continuous automatic record—would, he thought, be competent to clear up any outstanding questions there might be as to the endurance of ropes of different constructions, both up to the point of the first wire breaking, and afterwards, up to the complete destruction of the rope under the same conditions of loading and bending as subsisted at the time of the first failure of a single wire.

Dr. William H. Maw (Past-President) said the author had referred quite rightly to the unnecessary use of reverse bends in a number of cranes that were generally employed. He thought in a large number of instances this arose from retaining arrangements which were perfectly satisfactory with chains, but which were decidedly unsatisfactory with ropes. A very prominent case of that kind was in the arrangement of foundry cranes, and cranes serving steam-hammers. A very common arrangement in such cranes was to have chains led over the pulleys as shown in the sketch, Fig. 13 (page 730). There was in this arrangement a 90° bend in one direction, next a reverse bend of 180°, and finally a 90° bend in the same direction as the original bend. Some years ago, when he was going through Messrs. Tannett-Walker's Works at Leeds, Colonel

(Dr. William H. Maw.)
Tannett-Walker showed him a very simple alteration which he had made in the cranes of this class, which was deserving of being adopted more largely. With cranes using chains the original arrangement was quite satisfactory; when ropes were put in very severe wear was experienced. To reduce this, Colonel Tannett-Walker put two pulleys side by side on the same axis, as shown in

Fig. 13.

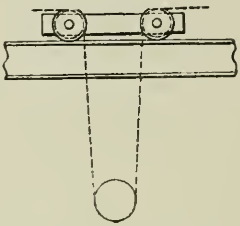


Fig. 14.

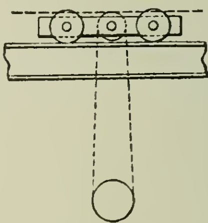


Fig. 14; he took the rope over one of these pulleys, then down to the hook pulley, round that pulley and then up over the second of the top pulleys. With that arrangement all the bends were in the same direction; and Colonel Tannett-Walker informed him that this simple alteration increased the life of the ropes, if he remembered rightly, about 50 per cent.

Mr. Thomas James said that one of the things which troubled him a little while ago was to what extent he could allow a rope on a crane to continue at work, assuming there were strands broken. He asked several engineers of experience what rule they followed in allowing a rope to continue at work after a certain number of strands had broken, presuming that very careful inspection was periodically made to see that there was no general deterioration in the rope; and he had some difficulty in getting any information from them. One engineer told him that he decided on the circumstances of the case at the time; another said that if it was not too bad, he would let it go on. Those seemed to him to be very unsatisfactory answers.

He had charge of a considerable amount of mechanical apparatus, and if he were asked why he allowed certain wire ropes

to continue working, he wanted to be in a position to give a satisfactory answer. He was able to refer to a few reliable tests, the results of which he would briefly quote to the Meeting. The first test was made on a rope of 2 inches circumference with six strands of 37 wires each, making 222 wires altogether, with hemp core. The basis upon which he set to work was that there should not be a factor of safety of less than 8 to 1. A rope, which had been in use for some time on that basis, was tested again and was found to be still within the margin, in fact it was rather curious that it came a decimal point or two over a piece of the same rope when new, namely, 8·23, as against 8·15, which was on record as its original factor of safety. Of that specimen two wires were subsequently broken, and the factor of safety still remained up to 8 to 1—actually 8·03, as compared with 8·23 with all the strands perfect. Continuing the experiment, 18 wires out of the 222 were cut through, and the factor of safety was then 7·5 to 1. Then an additional 20 wires were cut, that is, four in each of five strands, close together, in addition to 22 broken wires, and the factor of safety then fell to 7·42, that is, 42 wires cut or broken out of a total of 222, leaving the factor of safety 7·42 to 1. A piece out of the same length of rope was again tested, 72 of the wires being cut (12 in each strand) in addition to 1 wire already broken, the factor of safety falling to 7·125 to 1. One of the answers which he obtained from the engineers whom he consulted was that they would not let the rope be worked unless it appeared safe, and one would hardly let a rope go on working with 73 wires broken. As a final test, to bring it down to a much lower factor of safety, two out of the six strands were cut right through at opposite sides, in addition to 5 wires broken, making a total of 79 wires cut or broken, and a factor of safety was then obtained of only 5·34 to 1.

Those experiments led them to decide that the strength of the rope decreased proportionately with the number of broken wires, irrespective of whether the breaks occurred in the same strand, or were divided amongst the other strands. The object of the tests was to fix a necessary scrapping limit. It was rather a serious matter to scrap a rope too soon, and it was also a serious matter to

see a rope working with certain of its strands broken, if one did not know exactly how it would stand. It was suggested that the results obtained from the experiments allowed a loss of 10 per cent. to be incurred, that is 22 wires cut still left 7·4 factor of safety against the original 8·2. That was without making any further allowance for margin of safety, or "so much for luck," an expression that an engineer ought hardly to use. But if 10 per cent. was allowed on figures from experiment, so much must be allowed for——

The President: The personal factor.

Mr. James agreed with the President. So far as the experiments were concerned, the principle was adopted that wire ropes on cranes should not be worked when more than 4 per cent. of the wires were broken, all other things being satisfactory, with frequent periodical examinations to see that there was no general deterioration in the ropes.

The President inquired whether the rope on which the experiments were made was a new rope, or whether it had been already heavily worn.

Mr. James replied that the rope had been in use for fifteen months, until two of the wires were broken.

Mr. Thomas Burke (Belfast Harbour Board) said that a good many wire ropes were used at the Belfast Harbour Works, in connection with which a system had been adopted whereby when the rope came from the crane-maker it received a number. It was put on the crane, and after a certain time it was taken off. No rope was allowed to work for longer than four months, if continuously working, or a certain number of turns. When a rope had lifted 10,000 tons, it was taken off. It was a 3-inch rope working over a 22-inch pulley with a straight lift where possible. The factor of safety used was not 8 to 1, but nothing less than 10 to 1.

The President inquired whether the rope was scrapped after it had lifted 10,000 tons.

Mr. Burke replied in the affirmative. 10,000 tons was the maximum amount of work a rope was allowed to do. In the large 7-inch ropes a life of ten years was allowed.

Dr. H. S. Hele-Shaw (Member of Council) said there was one aspect of the question that had not been dealt with in the discussion, namely, the question of wear at the point of contact. If the ordinary lay was considered, in which the winding was taken across as in an ordinary rope, it would be seen that there was a great difference between the contact of the separate wires in this case and in the case in which Lang's lay was used. This resulted in a corresponding difference in the wear at the point of contact of a wire rope wound against the twist and a rope that was wound with the twist. It was known that Lang's lay gave trouble in connection with the rope as a whole, because it had not the property which kept the rope together, but it appeared to him to be a much better kind of rope from the point of view of contact wear. This was a point, however, on which some interesting questions arose. One thing was certain, that this contact wear for ropes running on pulleys needed investigation.

During the time that he was in South Africa a terrible accident occurred, in which forty Kaffirs were killed through the snapping of a rope, whereby they fell down the whole length of Robinson's deep mine—a distance of between three and four thousand feet. He was at the Technical Institute at Johannesburg then, and had the rope to examine. Taking a separate wire and stretching it out, he discovered that where it came in contact with the pulley it was worn through in many cases. All the wires that touched the pulley were worn at this point of contact. Why did that take place? There was no doubt, that what happened was that where the wire rope was bent round, the individual wire slipped on the pulley. There was not the slightest doubt that there must be a stretch on those large pulleys under the enormous strain, and

(Dr. H. S. Hele-Shaw.)

that the wire was not worn by pressing against the pulley but by slip in the actual take on and off owing to the stretch.

The accident in question practically led to the abandonment of the flat rope, though the flat rope appeared quite a feasible thing because it had so little bending strain. No doubt the reason why disasters occurred with flat ropes was because the ropes could not be inspected. The way to inspect an ordinary wire rope was to prise it open and examine the wires separately, but this could not be done with a flat rope. The suggestion he desired to make, in conclusion, was that the whole question of the wear of wire ropes over pulleys ought to be investigated by the Institution. He did not refer so much to the ropes that were used for direct pull, but to the ropes that were being put into use every day in ever-increasing numbers for driving over pulleys, and particularly in connection with lifts and hoists.

Mr. DANIEL ADAMSON, in reply, in the first place thanked the President for pointing out the great importance of care and inspection in the use of wire ropes. The designer and the wire-rope maker might each do his best, but by negligent use afterwards the rope could very quickly be seriously damaged.

Mr. Robert Matthews (page 723) had criticized his method of comparing the number of bends to which the rope was subjected when working under the different arrangements indicated by Figs. 5–12 (page 716) and perhaps some explanation on this point was desirable. He (the author) took into consideration only the number of bends through which a given length of rope could pass while the load travelled from the lowest to the highest point; for example, in Fig. 5 (mentioned particularly by Mr. Matthews) the rope was certainly bent in more than one place, but as no portion of the rope could pass through more than one of the bends while at work he described this arrangement as including one bend. Turning to Fig. 6 (also quoted by Mr. Matthews), it would be quite in the ordinary course of events for one portion of the rope to pass under the bottom pulley (making a bend at entering and a second bend on leaving) and then on to the barrel (making a third

bend), and accordingly he had described this as a three-bend arrangement. The bend at the compensating pulley was not included in any example, because that portion of the rope did not pass any other pulleys and was therefore unaffected by the arrangement adopted for the drum and pulleys.

Mr. Matthews had quoted his reference to the compensating pulley at the foot of page 721. His intention in inserting that paragraph was to emphasize the importance of making the compensating pulley of reasonable size. Although it was not so important as the working pulleys, yet it was a point that must not be overlooked in the design. Mr. Matthews referred to Lang's lay as against the ordinary lay, and that had also been referred to by some of the other speakers; and it was also mentioned in the last few words of the Paper. The information he had from the rope makers was that the Lang's lay must be more carefully dealt with in putting on and in working than ropes with ordinary lay.

Mr. Matthews also took exception to the quotation of some figures given on page 710. It was perhaps not necessary to explain that he did not intend to imply that Mr. Matthews did not work his cranes efficiently, but he quoted those figures, as the only published figures with which he was acquainted, to show the great extremes that took place in the working of cranes. If there were any other authoritative figures available, he was sure all the members would be pleased to hear of them. Personally he was trying to emphasize the fact that some cranes worked hundreds of times and others only tens of times in the course of a week. He gathered from Mr. Matthews' remarks that he confirmed the accuracy of the figures, although he objected to the deductions which might be drawn from them. Mr. Matthews suggested that 20 to 25 times the diameter of the rope was suitable for barrels and pulleys. On page 718 of the Paper he had suggested that 7 circumferences, or say 22 diameters, was a reasonable figure. If Mr. Matthews would insist upon 25 diameters, or say 8 circumferences, for his barrels, and would make the bottom pulleys 2 circumferences larger, his (the author's) opinion was that the ropes would last twice as long as

(Mr. Daniel Adamson.)
they did at present. That was a point made in the Paper—that it was important to increase the diameters of the bottom pulleys.

Mr. Matthews also suggested that a double bend was worse than two ordinary bends. In making that statement he (the author) merely quoted from Mr. Biggart's results. If a reverse bend was worse than two ordinary bends, there was the greater reason for designing cranes and hoists so as to avoid them. There was one statement of Mr. Matthews with which he agreed, namely, that he hoped the discussion would help all the members to come to a right conclusion.

Mr. A. Basil Wilson also referred (page 725) to the disadvantage of reverse bends, and pointed out that, in the building in which the Meeting was being held, a hoist with two reverse bends and three primary bends lasted one session, whereas when an arrangement, including only one primary bend, was substituted very beneficial results followed. According to the Paper the rope should last seven times as long. Life was short, and it would be necessary to wait a little while to see whether that was the case, that is, that two reverse bends and three primary bends were seven times as destructive to the rope as the single bend now in use.

Mr. Wilson asked a question as to the effect of partial bends. His own opinion was that partial bends would be quite as objectionable as whole bends, if the loading was similar in the two cases. The opinion that had generally been held up to the present was that the life of a rope depended upon the stresses due to bending, but it depended very largely upon the amount of the abrasion that took place, and the amount of the abrasion depended certainly upon the loading between the rope and the pulley—exactly to what extent he had not been able to ascertain. As a rough approximation, a quarter of the wear and tear on a rope might be put down to the fatigue due to bending, and three-quarters of it to the abrasion which took place between the wires and the pulleys. That was a very rough approximation, but it showed the importance of considering abrasion when discussing the life of a rope. He suggested that with a partial bend the abrasion

might be proportionate to the angle included in the arc of contact with the pulley as compared with a 180° bend.

Mr. Wicksteed referred (page 727) to the difference between the strength of single wires and the aggregate strength of the same wires when built up into a rope. Ropes never quite attained the full strength of the aggregate wires in the rope, for the reason that Mr. Wicksteed explained, namely, the difficulty of equal loading of the wires. Mr. Wicksteed also emphasized the importance of the attachment of the wire rope to the hoisting appliance, and that was certainly a matter which deserved consideration under the heading of the Paper.

Another point Mr. Wicksteed mentioned was that in deep mines the effects of fatigue appeared to be shown at the attachment, presumably at the attachment to the cage rather than to the drum. Failures would generally occur at the attachment of the rope to the cage, if the breakage was due to the sudden starting of the load, for the reason that the effects of inertia were very much more severely felt at that point.

Mr. Wicksteed mentioned a testing machine for investigating the durability of ropes, and Dr. Hele-Shaw had suggested (page 734) that a Committee of the Institution should institute a research. The President had said the subject required research, and he would like personally to recommend the question of the durability of wire ropes to the consideration of the Council as a desirable object for investigation.

Dr. Maw referred (page 729) to the very frequent fault in human affairs of retaining old arrangements with new devices, until one found out the error and the mistake in so doing. Dr. Maw had given an example of an advantageous application of the common sense which engineers were to a smaller or greater extent endowed with, and it rather seemed to be a reflection upon Mr. Wicksteed's suggestion that a machine could find out such things. There was room both for the machine and for the brain power of the individual.

(Mr. Daniel Adamson.)

Mr. Thomas James gave some very interesting figures (page 731) obtained from the testing of ropes which had been in use about two years, and in which the strands were experimentally cut through, and suggested that when 4 per cent. of the wires were broken the life of the rope was ended. It was very difficult to count the number of wires broken in a rope unless they all happened to be in the same place, because if they were distributed along the whole length of the rope it might be thought that each wire was only one out of the 300 odd and that the other wires were intact near it. The suggestion was not original—it had been suggested to him during the Meeting—that the whole of the wires, or a very large number of the wires, might be broken in different parts of the same rope, the first intimation of which was that the rope pulled out; and, due to that element of luck which one speaker referred to, it frequently happened that the rope pulled out when under a very much smaller load than it had been previously carrying.

*Communications.*

Mr. H. Lowthian Barge wrote that, dealing with Section (*a*), quality of material and size of wire, the author remarked that some rope-makers claimed as an advantage for the stronger material that "it does enable smaller pulleys to be used." The writer would like to point out that what was claimed was not that smaller pulleys could be used, but that ropes of greater ultimate breaking stress might be used for pulleys of the same diameter.

With regard to paragraph 2 on page 709, it would appear that the author was assuming that, in increasing the number of wires, the additional wires were always added in the outer circumference of the strands, as he said that in a rope having four times as many wires in it, the pressure between the wires and the sheaves might be assumed to be one-quarter the pressure in the original case. As a matter of fact, when the number of wires in a rope was increased,

in almost all cases these wires were introduced in the centre of the strands. Fig. 15, Plate 39, showed ropes having various numbers of wires in each strand, which he hoped would make this point clear.

With regard to the internal abrasion (page 710), it was very desirable in ordering crane or other running rope that it should be specified that the individual wires should be thoroughly lubricated during manufacture. By this means a considerable amount of abrasion was avoided, as in oiling the outside of a rope it was by no means likely that the oil would percolate through to the inner wires of the multiple wire strands.

With regard to the last paragraph in the Paper (page 722), ropes made on the so-called Lang's lay principle should be used with both ends fixed or spliced into an endless band, otherwise they had a tendency to unlay. Undoubtedly ropes made on Lang's lay presented more wearing surface, as could be readily seen from the illustrations at the foot of Plate 39, but size for size with ropes of ordinary lay they were not so flexible, as well as having the disadvantages mentioned above. He thought the Paper was most interesting, and that it added considerably to the somewhat meagre amount of technical information available with regard to the working of wire ropes.

Mr. H. H. Broughton wrote that he agreed, in the main, with many of the author's deductions. He was, however, of the opinion that numerical constants for the determination of the life of crane ropes based on Mr. Biggart's experiments were not as valuable to designers as they might have been, had the experiments taken into account several factors which had a far-reaching effect on the life of a rope. Three of the factors were:—(i) Frequency of starting and stopping; (ii) Effect of reversal; and (iii) Effect of impact. There were other factors that were not covered by Biggart's experiments.

He regarded with considerable suspicion the rule given in the Paper for doubling the life of a rope, and he would ask the author if his firm had succeeded in doubling the life of a rope in that way.

(Mr. H. H. Broughton.)

Perhaps he had drawn wrong conclusions, and to make certain he would like the author's opinion in a concrete case, namely:—that of a 50-ton crane with eight parts of rope, $3\frac{1}{8}$ inches in circumference, reeved double, and winding two parts on the barrel, block pulleys 22 inches in diameter. Was he correct in assuming that if the pulleys had been made $28\frac{1}{4}$ inches in diameter, the rope would have lasted twice as long?

It was his opinion that the author had not made it clear that very large proportions were necessary to ensure a reasonable life for ropes on cranes with many falls of rope. In the Table below he had set forth the summarized data for the lifting gears of five 50-ton cranes. The lifting speed was 10 feet per minute in each case, and the motors were each rated at 50 h.p. at 420 revolutions per minute.

| — | Number of parts of rope carrying the load. | | | | |
|---|---|---|---|---|---|
| | 4 | 6 | 8 | 10 | 12 |
| Force on each rope . . . tons | $12\frac{1}{2}$ | $8\frac{1}{3}$ | $6\frac{1}{4}$ | 5 | $4\frac{1}{6}$ |
| Force acting on barrel . . . ,, | 25 | $16\frac{2}{3}$ | $12\frac{1}{2}$ | 10 | $8\frac{1}{3}$ |
| Diameter of rope . . inches | $1\frac{3}{8}$ | $1\frac{1}{8}$ | 1 | $\frac{7}{8}$ | $\frac{3}{4}$ |
| Circumference of rope . . ,, | $4\frac{1}{4}$ | $3\frac{1}{2}$ | $3\frac{1}{8}$ | $2\frac{3}{4}$ | $2\frac{3}{8}$ |
| Diameter of barrel and sheaves ,, | 33 | 27 | 24 | 21 | 18 |
| Circumference of barrel and sheaves ft. | 8·64 | 7·07 | 6·28 | 5·5 | 4·71 |
| Speed of barrel . . revs. per min. | 2·32 | 4·25 | 6·37 | 9·1 | 12·74 |
| Speed reduction . . . . . | 181 | 99 | 66 | 46·2 | 33 |

He would take the two extremes, namely, the cases where the load was supported by four parts of rope, and by twelve parts of rope. Neither was a freak arrangement. According to the author, the diameter of the barrel and sheaves for the small rope ($2\frac{3}{8}$ inches in circumference) should be made nearly equal to the diameter for the large rope ($4\frac{1}{4}$ inches in circumference) if the ropes had to be

equally durable. Did the author suggest that with a $\frac{3}{4}$-inch rope and 18-inch sheaves, the rope would have only one-eighth the life of a $1\frac{3}{8}$-inch rope on 33-inch sheaves? He considered a $\frac{3}{4}$-inch rope and 18-inch sheaves to be a very liberal design, and with proper attention the rope would last for years.

On the question of rope lubrication he was in entire agreement with the author, and he thought that makers would be well advised to provide an appliance on the trolley for lubricating the rope. He agreed that compensating pulleys should be of large diameter, in order to minimize the effect of repeated bending of the rope at the same place due to the slight movement of the rope.

With regard to "Lang's lay" rope, his attention had recently been drawn to a rope in which the strands were double, one on the other, so arranged that each rope acted against the tendency of the other to twist, which ensured the rope remaining even without any tendency to spin.

As an advocate of the application of scientific principles to the design of crane mechanisms, he thought the Paper was a valuable addition to the literature on the subject. No one would deny that the question of rope life was important, but there were many other such questions which would have to be tackled before crane practice could be considered satisfactory.

Mr. GEORGE HUGHES (Member of Council) wrote that the Lancashire and Yorkshire Railway Company employed a large number of wire ropes, and the tabulated statement (pages 744–7) gave full particulars of some of them, of which he had every reason to believe that the observations were accurately made. He gave them as existing facts, leaving the members to conclude that the weak points would be rectified. These perhaps might be of greater service to the members than adopting a counsel of perfection.

On analysing the statement, it would be noted that the poorest life in number of lifts was Victoria Street Depot (Liverpool) goods lift, namely 9,396 lifts with a ratio of 36 and middle grade of steel and lowest flexibility, Lang's lay. The ratios of pulley diameters

(Mr. George Hughes.)

were good, namely, 36 and 47. On the face of it this should be one of the longest-lived ropes. The average lift was 13 cwt. against 30 cwt. capacity. This rope must have been injured although there was no record of it. The best life was Bolton goods hoist with 478,675 lifts. The pulley ratios were very good, namely, 36·5 and 47. Lang's lay, middle grade of steel and middle flexibility. It would be expected that this should also be good, but a lower grade of steel and a lower flexibility should suffice in the ordinary way. The next best was the fellow to this hoist on the other platform.

The next two examples were Liverpool, Victoria Street Depot, passenger hoist. Ratios 51 and 36. Lang's lay. Cheapest grade of steel. The next two were the ropes on the Halliwell electric 30-cwt. cranes. Ratios only 20·5. Middle grade steel, lowest flexibility, ordinary lay (old). The next two were cranes—one 5-ton at Newton Heath and the other 7-ton steam jib-crane at Wyre Dock. Neither of these was worn out yet. Ratios 24 and 14·5, the latter being very low indeed; it was a special compound rope known as "Nuflex." The Newton Heath crane is middle steel and lowest flexibility. The next three 9, 10, 11, are Newton Heath overhead cranes. Ratios 21 to 27. 3rd flexibility, middle steel. The average load per lift was very low indeed. Owing to their high flexibility the ropes were liable to injury.

Dealing with some of the others, namely, Werneth Cotton Shed overhead cranes, 15,794 and 23,700 lifts. These were low and their ratios were 19·8. Lowest grade steel. Ordinary lay and lowest flexibility. For this ratio and ordinary lay a higher flexibility and a middle grade steel should give a longer life. Wyre Dock transporter, Fig. 27 (page 743), 40,491 lifts, which was very poor. The ratio, 18·8, was bad, and there was a treble turn. The wear was excessive and new pulleys were being provided to overcome this trouble. Compound ropes. Lowest grade steel.

The two cranes in the heavy machine shop at Horwich gave a low life to their ropes. They had a middle flexibility and middle quality of steel, with 19·4 ratio, which was evidently the trouble. The ropes were large and should have 26–30 ratios. The top one in the list, namely, Horwich stores yard 30-cwt. electric crane, was bad

*Examples of Pulley Arrangements for Wire Ropes on Cranes and Hoists of the Lancashire and Yorkshire Railway Company.* (*Figs.* 18–31 *are referred to in Table, pp.* 744–7.)

Fig. 16.

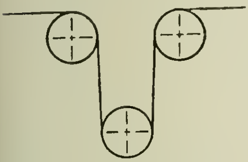


Fig. 17.

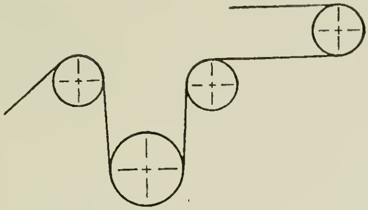


Fig. 18.

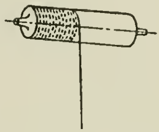


Fig. 19.

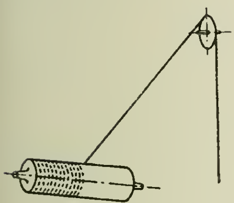


Fig. 20.

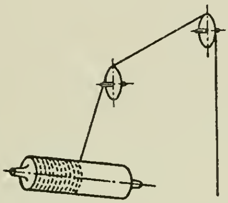


Fig. 21.

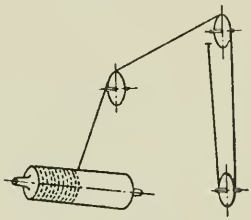


Fig. 22.

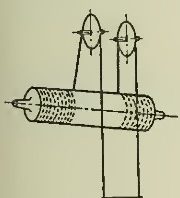


Fig. 23.

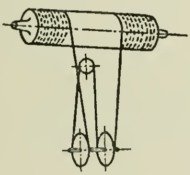


Fig. 24.

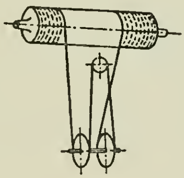


Fig. 25.

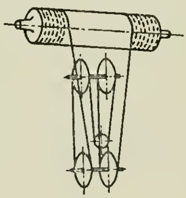


Fig. 26.

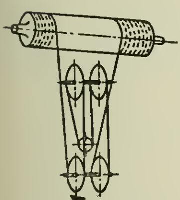


Fig. 27.

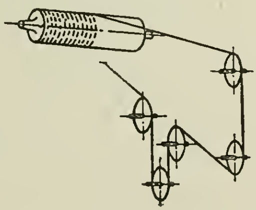


Fig. 28.

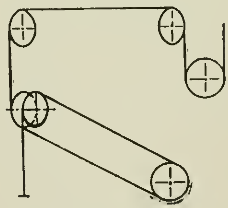


Fig. 29.

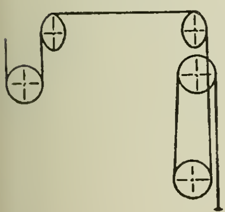


Fig. 30.

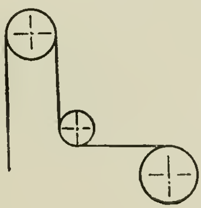
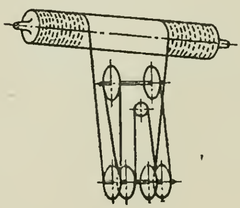


Fig. 31.



(Mr. George Hughes.)

TABLE 1 (*continued on opposite page*).

*Particulars of Work done by Wire Ropes on Specific Cranes and Hoists on L. & Y. Railway during the month of May* 1912.

Arranged in Order of Merit.

| | Fig. (p. 743). | Place. | Appliance. | Registered Capacity. | | Maker of Lifting Appliances. |
|---|---|---|---|---|---|---|
| | | | | t. | c. | |
| 1 | 19 | Bolton Passenger, Down Platform | Electric Goods Hoist | 1 | 10 | Waygood |
| 2 | 19 | ,, ,, Up Platform | ,, ,, ,, | 1 | 10 | ,, |
| 3 | 30 | Liverpool, Victoria Street Depot | Electric Passenger Hoist | 0 | 7 | ,, |
| 4 | 30 | ,, ,, ,, ,, | ,, ,, ,, | 0 | 7 | ,, |
| 5 | 18 | Halliwell Goods Warehouse . | Electric Overhead Crane | 1 | 10 | Crompton |
| 6 | 18 | ,, ,, ,, . | ,, ,, ,, | 1 | 10 | ,, |
| 7 | 20 | Wyre Dock . . . . | Steam Road Crane | 4 | 0 | Isles |
| 8 | 24 | Newton Heath Machine Shop . | Electric Overhead Crane | 5 | 0 | Heywood & Co. |
| 9 | 24 | ,, ,, Lifting Shed, W. | ,, ,, ,, | 10 | 0 | J. Berry |
| 10 | 24 | ,, ,, ,, ,, E. | ,, ,, ,, | 10 | 0 | ,, |
| 11 | 23 | ,, ,, Saw Mill . . | ,, ,, ,, | 5 | 0 | Craven Bros. |
| 12 | 19 | Bolton Goods Warehouse . . | Electric Jib Crane | 2 | 10 | P. R. Jackson |
| 13 | 19 | ,, ,, ,, . | ,, ,, ,, | 1 | 10 | ,, |
| 14 | 24 | Horwich Millwrights' Shop . | Electric Overhead Crane | 10 | 0 | J. Berry |
| 15 | 26 | Bolton Goods . . . . | Electric Goliath | 40 | 0 | A. Chaplin |
| 16 | 24 | Horwich Millwrights' Shop . | Electric Overhead Crane | 10 | 0 | J. Berry |
| 17 | 21 | Wyre Dock . . . . . | Steam Road Crane | 7 | 0 | Isles |
| 18 | 23 | North Mersey . . . . . | Electric Cantilever Crane | 10 | 0 | Musker |
| 19 | 26 | Horwich Millwrights' Shop . | Electric Overhead Crane | 20 | 0 | J. Berry |
| 20 | 28 | Goole Railway Dock . . | Hydraulic Coal Tip Table Tipping | 40 | 0 | Tannett-Walker |
| 21 | 27 | Wyre Dock . . . . . | Electric Transporter | 1 | 5 | Cowans Sheldon |
| 22 | 23 | Horwich Heavy Machine Shop . | Electric Overhead Crane | 10 | 0 | Vaughan |
| 23 | 20 | Wyre Dock . . . . . | Steam Road Crane | 4 | 0 | Isles |
| 24 | 23 | Horwich Heavy Machine Shop . | Electric Overhead Crane | 10 | 0 | Vaughan |
| 25 | 19 | ,, Stores Yard . . | Electric Jib Crane | 1 | 10 | P. R. Jackson |
| 26 | 18 | Werneth Cotton Shed . . | Electric Overhead Crane | 0 | 15 | Spencer & Co. |
| 27 | 31 | Newton Heath Saw-Mill Gantry | ,, ,, ,, | 7 | 10 | Heywood & Co. |
| 28 | 18 | Werneth Cotton Shed . . | ,, ,, ,, | 0 | 15 | Spencer & Co. |
| 29 | 22 | Liverpool, Victoria Street Depot | Electric Goods Hoist | 1 | 10 | Waygood |
| 30 | 25 | Horwich Boiler Shop . . | Electric Gantry | 15 heavy | | L. & Y. Railway |
| 31 | 29 | Goole Railway Dock . . | Hydraulic Coal Tip (Hopper Tipping) | 40 | 0 | Tannett-Walker |
| 32 | 19 | Bolton Goods Warehouse . . | Electric Goods Hoist | 1 | 10 | P. R. Jackson |

(*continued on next page*) TABLE 1.

*Particulars of Work done by Wire Ropes on Specific Cranes and Hoists on L. & Y. Railway during the month of May* 1912.

Arranged in Order of Merit.

| | Circumference of Rope. | Class of Steel. | Construction. | No. of Strands, and Wires in each Strand. | | Diameter of Barrel. | Diameter of Jib-Head Pulley. | Diameter of Bottom Pulley. |
|---|---|---|---|---|---|---|---|---|
| | inches. | | | strands. | wires. | inches. | inches. | inches. |
| 1 | 2 | Improved Plough | Lang's Lay | 6 | 27 | $23\frac{1}{4}$ | — | — |
| 2 | 2 | ,, ,, | ,, | 6 | 27 | $23\frac{1}{4}$ | — | — |
| 3 | 2 | Improved Crucible | ,, | 6 | 12 | $32\frac{1}{2}$ | — | — |
| 4 | 2 | ,, ,, | ,, | 6 | 12 | $32\frac{1}{2}$ | — | — |
| 5 | $1\frac{3}{4}$ | Improved Plough | old | 6 | 19 | $11\frac{3}{8}$ | — | — |
| 6 | $1\frac{3}{4}$ | ,, ,, | ,, | 6 | 19 | $11\frac{3}{8}$ | — | — |
| 7 | $3\frac{1}{4}$ | Improved Crucible | Compound | 34 | 7 | 15 | 24 | — |
| 8 | $1\frac{1}{2}$ | Improved Plough | Ordinary | 6 | 19 | $11\frac{1}{2}$ | — | 14 |
| 9 | 2 | ,, ,, | ,, | 6 | 30 | 13 | — | 13 |
| 10 | 2 | ,, ,, | ,, | 6 | 30 | 13 | — | 13 |
| 11 | $1\frac{1}{2}$ | ,, ,, | ,, | 6 | 19 | $13\frac{1}{8}$ | — | $10\frac{1}{2}$ |
| 12 | $2\frac{1}{4}$ | ,, ,, | old | 6 | 37 | $14\frac{5}{8}$ | $14\frac{3}{8}$ | — |
| 13 | $1\frac{5}{8}$ | ,, ,, | ,, | 6 | 14 | $13\frac{1}{2}$ | $11\frac{3}{8}$ | — |
| 14 | $1\frac{7}{8}$ | ,, ,, | Ordinary | 6 | 19 | 13 | — | $13\frac{3}{4}$ |
| 15 | $3\frac{7}{8}$ | ,, ,, | Lang's Lay | 6 | 37 | 23 | — | 21 |
| 16 | $1\frac{7}{8}$ | ,, ,, | Ordinary | 6 | 19 | 13 | — | $13\frac{3}{4}$ |
| 17 | $2\frac{3}{4}$ | ,, ,, | Compound | 34 | 7 | 18 | 21 | 17 |
| 18 | $2\frac{1}{4}$ | ,, ,, | Lang's Lay | 6 | 27 | $17\frac{1}{4}$ | — | $17\frac{1}{4}$ |
| 19 | $2\frac{1}{4}$ | ,, ,, | Ordinary | 6 | 27 | $14\frac{1}{2}$ | — | $17\frac{7}{8}$ |
| 20 | $3\frac{3}{4}$ | ,, ,, | Compound | 34 | 7 | 36 | — | — |
| 21 | 2 | Crucible | ,, | 17 | 7 | 24 | — | 12 |
| 22 | 2 | Improved Plough | Ordinary | 6 | 27 | 15 | — | $12\frac{3}{8}$ |
| 23 | $3\frac{1}{4}$ | Improved Crucible | ,, | 34 | 7 | 15 | 24 | — |
| 24 | 2 | Improved Plough | ,, | 6 | 27 | 15 | — | $12\frac{3}{8}$ |
| 25 | $1\frac{5}{8}$ | ,, ,, | ,, | 6 | 19 | $11\frac{3}{4}$ | 12 | — |
| 26 | $1\frac{5}{8}$ | Improved Crucible | old | 6 | 14 | $10\frac{1}{4}$ | — | — |
| 27 | $1\frac{1}{2}$ | Improved Plough | Ordinary | 6 | 19 | 18 | — | 12 |
| 28 | $1\frac{5}{8}$ | Improved Crucible | old | 6 | 14 | $10\frac{1}{4}$ | — | — |
| 29 | 2 | Improved Plough | Lang's Lay | 6 | 19 | 23 | — | — |
| 30 | 2 | ,, ,, | Ordinary | 6 | 27 | $20\frac{3}{8}$ | — | $18\frac{1}{8}$ |
| 31 | $4\frac{1}{4}$ | ,, ,, | Compound | 34 | 7 | 42 | — | — |
| 32 | $2\frac{1}{4}$ | ,, ,, | Lang's Lay | 6 | 37 | $35\frac{1}{2}$ | — | — |

(Mr. George Hughes.)

TABLE 1 (*continued from previous page*).

*Particulars of Work done by Wire Ropes on Specific Cranes and Hoists on L. & Y. Railway during the month of May* 1912.

Arranged in Order of Merit.

| | Fig. (p. 743). | Place. | Appliance. | Diameter of Guide or Jockey Pulley. | Days Worked. | Lifts Made. (*a*) |
|---|---|---|---|---|---|---|
| | | | | inches. | | |
| 1 | 19 | Bolton Passenger, Down Platform | Electric Goods Hoist | $29\frac{3}{4}$ | 27 | 12,337 |
| 2 | 19 | ,, ,, Up Platform | ,, ,, ,, | $29\frac{3}{4}$ | 27 | 8,187 |
| 3 | 30 | Liverpool, Victoria Street Depot | Electric Passenger Hoist | 23 | 26 | 2,603 |
| 4 | 30 | ,, ,, ,, ,, | ,, ,, ,, | 23 | 26 | 2,563 |
| 5 | 18 | Halliwell Goods Warehouse . | Electric Overhead Crane | — | 27 | 9,450 |
| 6 | 18 | ,, ,, ,, . | ,, ,, ,, | — | 27 | 10,974 |
| 7 | 20 | Wyre Dock . . . . | Steam Road Crane | 21 | 17 | 6,475 |
| 8 | 24 | Newton Heath Machine Shop . | Electric Overhead Crane | — | 25 | 8,564 |
| 9 | 24 | ,, ,, Lifting Shed, W. | ,, ,, ,, | — | 26 | 3,341 |
| 10 | 24 | ,, ,, ,, ,, E. | ,, ,, ,, | — | 26 | 6,481 |
| 11 | 23 | ,, ,, Saw Mill . . | ,, ,, ,, | — | 25 | 3,628 |
| 12 | 19 | Bolton Goods Warehouse . . | Electric Jib Crane | — | 27 | 2,972 |
| 13 | 19 | ,, ,, ,, . . | ,, ,, ,, | — | 27 | 3,877 |
| 14 | 24 | Horwich Millwrights' Shop . | Electric Overhead Crane | — | 27 | 993 |
| 15 | 26 | Bolton Goods . . . . | Electric Goliath | — | 27 | 2,687 |
| 16 | 24 | Horwich Millwrights' Shop . | Electric Overhead Crane | — | 27 | 955 |
| 17 | 21 | Wyre Dock . . . . | Steam Road Crane | 17 | 13 | 1,422 |
| 18 | 23 | North Mersey . . . . | Electric Cantilever Crane | — | 26 | 2,422 |
| 19 | 26 | Horwich Millwrights' Shop . | Electric Overhead Crane | — | 27 | 650 |
| 20 | 28 | Goole Railway Dock . . . | Hydraulic Coal Tip | — | 25 | 2,626 |
| 21 | 27 | Wyre Dock . . . . | Electric Transporter | 12 | 25 | 8,998 |
| 22 | 23 | Horwich Heavy Machine Shop . | Electric Overhead Crane | — | 27 | 4,016 |
| 23 | 20 | Wyre Dock . . . . | Steam Road Crane | 21 | 13 | 1,408 |
| 24 | 23 | Horwich Heavy Machine Shop . | Electric Overhead Crane | — | 27 | 1,880 |
| 25 | 19 | ,, Stores Yard . . | Electric Jib Crane | — | 27 | 527 |
| 26 | 18 | Werneth Cotton Shed . . | Electric Overhead Crane | — | 26 | 7,900 |
| 27 | 31 | Newton Heath Saw Mill Gantry | ,, ,, ,, | — | 24 | 3,251 |
| 28 | 18 | Werneth Cotton Shed . . | ,, ,, ,, | — | 26 | 7,897 |
| 29 | 22 | Liverpool, Victoria Street Depot | Electric Goods Hoist | 30 | 26 | 87 |
| 30 | 25 | Horwich Boiler Shop . . | Electric Gantry | — | 27 | 876 |
| 31 | 29 | Goole Railway Dock . . | Hydraulic Coal Tip | — | 20 | 50 |
| 32 | 19 | Bolton Goods Warehouse . . | Electric Goods Hoist | 16 | 27 | 2,171 |

(*concluded from page* 744) TABLE 1.

*Particulars of Work done by Wire Ropes on Specific Cranes and Hoists on L. & Y. Railway during the month of May* 1912.

Arranged in Order of Merit.

| | Average Load per Lift. (*b*) | | Average Height Lifted. | Maximum Load. | | Minimum Load. | | Life of Rope Already Renewed. | Life of Rope in Number Lifts of based on (*a*). | Life of Rope in Tons Handled per Rope. | Ratio of Barrel Diameter to Rope Diameter. | Ratio of Bottom or Jockey Pulley Diameter to Rope Diameter. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | t. | c. | feet. | t. | c. | t. | c. | months. | | | diameters. | diameters. |
| 1 | 0 | 4½ | 17·5 | 1 | 8 | 0 | 1 | 38·8 | 478,675 | 108,180 | 36·5 | 47 |
| 2 | 0 | 5·1 | 17·4 | 1 | 8 | 0 | 1 | 45·5 | 372,508 | 95,548 | 36·5 | 47 |
| 3 | 0 | 4½ | 90 | 0 | 7 | 0 | 1 | 27 | 304,551 | 68,521 | 51 | 36 |
| 4 | 0 | 4½ | 90 | 0 | 7 | 0 | 1 | 27 | 299,871 | 67,471 | 51 | 36 |
| 5 | 0 | 5·9 | 12 | 1 | 8 | 0 | 0¾ | 26·25 | 248,062 | 72,680 | 20·5 | — |
| 6 | 0 | 5·4 | 6·7 | 1 | 8 | 0 | 1 | 21 | 230,454 | 62,107 | 20·5 | — |
| 7 | 0 | 9·1 | 17·2 | 3 | 0 | 0 | 3 | 27* | 174,825 | 79,545 | 14·5 | 23·2 |
| 8 | 0 | 6¾ | 6 | 5 | 0 | 0 | 1 | 17* | 145,588 | 48,962 | 24·1 | 29·4 |
| 9 | 0 | 7 | 5·1 | 5 | 0 | 0 | 1 | 42 | 140,322 | 49,604 | 21 | 20·4 |
| 10 | 1 | 11 | 10·3 | 10 | 0 | 0 | 1 | 18 | 116,658 | 123,135 | 21 | 20·4 |
| 11 | 0 | 12½ | 13·5 | 5 | 0 | 0 | 1 | 30 | 108,840 | 68,400 | 27·5 | 22 |
| 12 | 0 | 4·6 | 5·4 | 1 | 6 | 0 | 1 | 32·3 | 106,000 | 24,576 | 20·4 | 20 |
| 13 | 0 | 4·6 | 5·6 | 10 | 0 | 0 | 1 | 20·8 | 80,641 | 18,547 | 26 | 22 |
| 14 | 0 | 5 | 4 | 5 | 0 | 0 | 0¼ | 74* | 73,482 | 18,370 | 21·8 | 23 |
| 15 | 1 | 2 | 7·2 | 20 | 0 | 0 | 1 | 26·7 | 71,877 | 79,208 | 18·6 | 17 |
| 16 | 0 | 5¼ | 4 | 5 | 0 | 0 | 0½ | 74* | 70,670 | 18,551 | 21·8 | 23 |
| 17 | 0 | 17·1 | 15·5 | 6 | 0 | 0 | 4 | 48 | 68,256 | 58,700 | 20·6 | 19·5 |
| 18 | 1 | 9 | 5·7 | 9 | 15 | 0 | 2 | 24 | 58,128 | 84,285 | 24 | 24 |
| 19 | 0 | 5½ | 3·7 | 4 | 0 | 0 | 0½ | 74* | 48,100 | 13,227 | 20·2 | 25 |
| 20 | 17 | 10 | 30 | 29 | 18 | 10 | 9 | 18 | 47,268 | 827,190 | 30 | None |
| 21 | 0 | 17½ | 21·5 | 1 | 5 | 0 | 0½ | 4·5 | 40,491 | 35,429 | 37·7 | 18·8 |
| 22 | 1 | 8 | 14 | 10 | 0 | 0 | 2¼ | 10 | 40,160 | 56,880 | 23·6 | 19·4 |
| 23 | 0 | 10·3 | 14 | 3 | 0 | 0 | 4 | 25* | 35,200 | 18,128 | 14·5 | 23·2 |
| 24 | 0 | 19 | 14 | 10 | 0 | 0 | 2¼ | 17 | 31,960 | 30,362 | 23·6 | 19·4 |
| 25 | 0 | 8 | 6 | 1 | 3 | 0 | 2 | 54 | 28,358 | 11,343 | 22·7 | 23·4 |
| 26 | 0 | 4½ | 12 | 0 | 6¾ | 0 | 3 | 3 | 23,700 | 5,334 | 19·8 | None |
| 27 | 0 | 7¾ | 6 | 5 | 0 | 0 | 1½ | 5* | 16,255 | 6,397 | 38 | 25 |
| 28 | 0 | 4½ | 12 | 0 | 6¾ | 0 | 3 | 2 | 15,794 | 3,553 | 19·8 | — |
| 29 | 0 | 13 | 25 | 1 | 5 | 0 | 4 | 108 | 9,396 | 6,107 | 36 | 47 |
| 30 | 1 | 15 | 6 | 11 | 6 | 0 | 0½ | 6* | 5,256 | 9,198 | 32 | 28·5 |
| 31 | 17 | 7 | 38 | 29 | 1 | 4 | 11 | 60 | 3,000 | 52,050 | 31 | None |
| 32 | 0 | 10·8 | 25 | 1 | 8 | 0 | 1 | 23·1 | 2,508 | 27,236 | 49·5 | 22·3 |

* First rope still in service.

(Mr. George Hughes.)

(28,358 lifts), for which there seemed no reason. Ratio 22·7, middle grade steel, lowest flexibility, ordinary lay. There should be a larger pulley and barrel. North Mersey cantilever was low lived. Ratio 24, middle flexibility, middle grade steel, Lang's lay. Wyre Dock steam cranes were bad. The ratio was low (14·5 and 20·6), although one crane was three and five times better than the other. They gave out where fastened to the socket with lead, owing to galvanic action.

All wire ropes were kept well greased or oiled, and galvanized ropes were not used. Their renewals were based upon the life of the rope preceding the present rope, except where the rope had had an accident. In a few cases it was the average of several renewals. Lang's lay could only be used in hoists where the ends were both fixed. Otherwise the ropes would unravel. Ordinary lay was less liable to unravel, but the wear on the rope was greater because a lesser surface of the wires lay upon the pulley. They tried to overcome this by using compound or "Nuflex" ropes, but the objection to these was their delicacy; they were easily injured by rubbing against anything, or a blow, or one coil lapping on another. The most surprising part of the report was the good work done by the Halliwell crane ropes, which were very hard worked and had a ratio of only 20·5 with a loose end.

The conclusions the writer drew were as follows:—

1. In general, use the lowest grade of steel.

2. In general, use the middle flexibility.

3. Use Lang's lay, if possible.

4. Ratio of pulley to rope diameter not less than 26.

5. 30 was better if it could be obtained.

6. If 30 to 40 could be got, use it and keep then to the lowest flexibility instead of middle flexibility.

7. Try to avoid special kinds of ropes.

8. If queer double turns were necessary like those shown in Fig. 17 (page 743), then let there be scarcely any limit to the pulley diameters, and have ratio to rope diameters, say 45 to 50, if at all possible. Only in this direction could reasonable length of life be secured.

Mr. C. HUMPHREY WINGFIELD wrote that, although much good work had been achieved in showing how experimental results applied to practical design, much still remained to be done. Mr. Adamson's Paper was a most valuable contribution to practical science in this respect. While fully agreeing with the emphasis which the author laid on the necessity for lubrication and the bad effect of reverse bends on wire ropes for cranes, he (Mr. Wingfield) found it difficult to follow the reasoning on pages 708–9, by which the author arrived at the result that of two ropes of equal diameter, sectional area, and load, one of which was made with wires half the diameter, and therefore four times as numerous as those in the other, the finer wires would wear out in half the time that the coarser ones would do if used on the same sized pulley, and in one quarter of the time if the diameter of the pulley bore the same proportion to that of the wires in each case. These ropes were usually made in six strands, and, if embraced by a closely fitting semicircular groove in a pulley, the maximum number of wires in contact with the groove at any section of an ordinary crane rope would be about six. Reference to the sectional views of flexible wire ropes given in certain makers' catalogues, indicated that with wires of half the usual diameter as many as nine might touch the pulley, if the diameter of the rope were the same as before. Thus the number of points of contact at a given section would be nine-sixths or only 50 per cent. more with the finer wire.

He (the writer) might be wrong, but he thought the author (page 709) had overlooked the fact that the pressure of individual wires on the pulley was not solely due to the tension in those particular wires, but also to the squeezing effect of the outer wires. Thus the total pressure on the pulley would be identical in the two cases, and the pressure per wire in the special rope would be six-ninths or two-thirds of that in the case of the ordinary rope. The author said it would be one-fourth. The pressure per square inch would vary inversely as the diameter of the wire and of the length in contact with the pulley (of which the latter's diameter was a measure). Hence, for the fine wires it would be $(\frac{2}{3} \div \frac{1}{2}) = \frac{4}{3}$ of that of the coarser wires if used on the large pulley and $\{\frac{2}{3} \div (\frac{1}{2} \times \frac{1}{2})\} = \frac{8}{3}$ in the case of the smaller pulley.

(Mr. C. Humphrey Wingfield.)

The rate of wear might reasonably be expected to bear some relation to the relative motion per inch of wires near the pulley, and, so far as bending alone was concerned, this would vary as the ratio of the diameter of the wire to that of the pulley.* Hence the rate of wear would perhaps be found to vary as this motion multiplied by the pressure per square inch against the sides of the wire, and the time required for the rope to wear out would then vary as the reciprocal of this multiplied by the diameter of the wire. If so, the fine wire on a larger pulley should last three-quarters as long as the coarser wire, and on a smaller pulley only three-sixteenths as long.

The following Table showed these calculations in detail.

| | | | | |
|---|---|---|---|---|
| Diameter of wire . . . . ($d$) | 1 | $\frac{1}{2}$ | | $\frac{1}{2}$ |
| Diameter of pulley . . . (D) | 1 | 1 | | $\frac{1}{2}$ |
| Total pressure on pulley . . P | 1 | 1 | | 1 |
| No. of wires in contact with pulley. N | 6 | 9 | | 9 |
| Pressure, per wire, on pulley $\left(\frac{6}{\mathrm{N}}\right)$ . . | 1 | $\frac{2}{3}$ | | $\frac{2}{3}$ |
| Pressure on wire per sq. in. $\frac{6}{d\,\mathrm{ND}}$ . ($p$) | 1 | $(\frac{2}{3} \div \frac{1}{2}) = \frac{4}{3}$ | | $\left\{\frac{2}{3} \div (\frac{1}{2} \times \frac{1}{2}) = \frac{8}{3}\right.$ |
| Amount of sliding per inch of wire $\frac{d}{\mathrm{D}}$ (S) | 1 | $\frac{1}{2}$ | | 1 |
| †Rate of wear $\frac{6}{\mathrm{ND}^2}$ . . . (W) | 1 | $\frac{2}{3}$ | | $\left\{\frac{2}{3} \div (\frac{1}{2} \times \frac{1}{2}) = \frac{8}{3}\right.$ |
| Time to wear to centre of wire $\left(\propto \frac{d}{\mathrm{W}}\right)$ (T) | 1 | $\frac{1}{2} \div \frac{2}{3} = \frac{3}{4}$ | | $\frac{1}{2} \div \frac{8}{3} = \frac{3}{16}$ |

The writer did not put these figures forward as of any value beyond indicating the direction which further experiments should take. So far as possible he had taken the figures for the thicker wire as of unit value.

* The total relative movement, in one direction, between two contiguous wires at different radii was twice as great when two bends in opposite directions occurred as when the rope bent in one direction only.

$$\dagger\ p \times \mathrm{S} = \left(\frac{6}{d\,\mathrm{ND}} \times \frac{d}{\mathrm{D}}\right) = \frac{6}{\mathrm{ND}^2}.$$

The author (page 709) rather despaired of any agreement being arrived at as to the effect of stress apart from wear, on the endurance by metals of repeated applications of load. During the discussion of Messrs. Eden, Rose, and Cunningham's Paper, the writer exhibited diagrams * which showed that, when allowance was made for the varying ultimate strengths of the materials, the experiments of different investigators were more consistent than might perhaps have been hoped for. He had been unable to conjecture why the author suggested that a reduction of from 40 to 25 tons range should increase the life 500 times. However, from Fig. 56 (in loc. cit.) he would have thought about $\frac{6000}{150}$, or say 40 times, would be a nearer approximation. He fully agreed, however, that even this figure was not likely to be approached in practice, and that abrasion was a more serious factor in the determination of the life of a rope.

The writer presumed the author did not attach much importance to Fig. 2 (page 713), as several of the lines seemed to depart rather widely from the dots through which the lines in Fig. 1 were drawn. He had checked several of the dotted lines in Fig. 3 (page 715), and found that they were accurately drawn through the experimental dots, so that the author's very simple and practical rule (page 714) was reliable to the extent shown by the divergence of the full and dotted lines in that Figure.

The question of the stress in a rope, subjected to bending as well as to tension, had not been satisfactorily determined. In Fig. 56 (already referred to) the writer had laid down the lowest line but one of the author's Fig. 1 (page 712), calculating the stress on the usual assumption; namely that the effect of bending in producing stress on the wire was the same as if it were parallel with the axis of the rope, and that the same applied to the effect of the longitudinal pull on the rope, the stresses given in the diagram being the sum of these two results. Obviously, owing to the twisted form of the rope, a torsional stress was produced on the wire by

* Proceedings 1911, Part 4, Figs. 55 and 56 (pages 905–6).

(Mr. C. Humphrey Wingfield.)

either bending or pulling the rope. Professor Unwin in his last edition of "Machine Design" had given, somewhat tentatively, a formula for making a correction on this account, but it was probably not worth while to complicate the calculations in this way when merely requiring relative results, as in this case. The friction on the pulley might very possibly have a still more serious effect, however, since instead of bending about its neutral axis there would be a tendency for the rope to bend about an axis nearer the pulley, as the sliding necessary for the first assumption to be true would be checked.

With regard to the last paragraph in the Paper (page 722), he understood that, for such purposes as hoisting men and material in the mines of the Transvaal, "Lang's lay" was used in preference to nearly every other make. Of course in such cases the cage was guided, which bore out what the author said as to this rope being most suitable in cases where spin was prevented.

The great importance of the points in design to which the author drew attention in Figs. 4 to 12 (page 716) and in the following pages, was very apparent after consideration of the facts which he demonstrated, and reference to catalogues of crane-makers showed that they had hitherto escaped notice. He trusted the discussion would be found worthy of the time and trouble such a Paper must obviously have taken to produce, and that the generosity with which Mr. Adamson had placed his results at the disposal of rival makers would be fully recognized. The writer wished to suggest that information should be given by rope-makers as to the behaviour of the hemp packing. To what extent was it efficient in preserving the strands in their original positions after a year or two of wear?

With regard to the suggestion made during the discussion that this subject was a matter well adapted for experimental examination, he (Mr. Wingfield) would like most heartily to support this proposal.

Mr. Daniel Adamson wrote, in reply to Mr. H. Lowthian Barge (page 738) that he (the author) would quote the actual paragraph from a ropemaker's letter upon which the statement referred to

was based: "This wire (115 tons per square inch) allows of a fairly small rope being used, and the advantage of this is that the working parts of the crane can be made smaller, which decreases the cost of the crane, though perhaps the rope might be somewhat higher in price than if it were made from steel of 90 tons per square inch." Mr. Barge made what appeared to be the same claim in different words, in that he would use pulleys of the same diameter for ropes carrying greater loads. Whatever ropemakers might say, there was a very general feeling amongst rope-users that a low tensile strength was preferable (*see* Mr. Hughes' Communication, page 748).

Regarding the number of wires in contact with the pulley with ropes of fine wires, as compared with ropes of coarse wires, the author was aware that grave doubts existed as to his estimate of four times the number, but if the increase were less than this, then the pressure per wire would be increased and the rope of fine wires would be shown (as it was known to be) even less durable.

In reply to Mr. H. H. Broughton (page 739), the author was sorry he could not yet say that the life of any ropes had been doubled by increasing the diameter of the bottom pulleys, because of the length of time involved by the question, but he could say that since he had been acquainted with the deductions to be made from Mr. Biggart's experiments he had given his own designs the benefit of them. The concrete case stated by Mr. Broughton of a 50-ton crane with eight parts of $3\frac{1}{8}$ inches circumference, rope and block pulleys 22 inches diameter, seemed to provide for a very low factor of safety (only about 6·6 as compared with the usually accepted minimum of 8); the ratio of rope circumference to pulley diameter (seven to one) was therefore illusory. Assuming a reasonable factor of safety and accepting the conclusions in the Paper under discussion, the rope ought to be $3\frac{1}{2}$ inches circumference and the barrel and pulleys 30 to 33 inches diameter, to give a life equal to a four-part arrangement of 5 inches circumference rope as Fig. 6 (page 716), with barrel and pulleys of the same diameter, that is, 30 to 33 inches. Answering Mr. Broughton's direct question, the

(Mr. Daniel Adamson.)
author believed that increasing the diameter of the pulleys, Fig. 12 (page 716), would lengthen the life of the rope by 50 per cent. (as stated on page 720). Whether the life of the rope would be doubled depended upon the average height of lift (as explained on page 721).

Replying to Mr. Broughton's other question regarding the proportions given in the Table (page 740), the author considered that larger ropes should be adopted throughout, and further that the ratio of pulley to rope diameter—about 22 to 1 in the list—should be increased to 33 to 1 for the design with 12 parts of rope to ensure reasonable durability (*see* Table 3, page 718). If the ratio were retained at 22 to 1 throughout the series, the life of the rope with twelve parts would only be one-quarter of that of the rope with four parts, according to Table 1 (page 717).

Mr. Broughton asked whether the life of a $\frac{3}{4}$-inch diameter rope on 18-inch sheaves would be one-eighth of that of a $1\frac{3}{8}$-inch diameter rope on 33-inch sheaves, but such a suggestion was not contained in the Paper, and the author thought Mr. Broughton must have misread page 721 which referred to a special advantage of Fig. 12, where large pulleys were used on a four-part rope. This advantage would not apply to a twelve-part arrangement, because the portion of the rope which passed under the lower pulleys would in this case soon reach the upper drum and be subjected to the further bends there. The author quite agreed with Mr. Broughton's closing remarks as to the numerous questions in crane design that awaited ventilation, and hoped that other members would take them up in the near future.

The members generally would feel much indebted to Mr. George Hughes for the very complete tabulation of experiences with wire-lifting ropes that he had contributed to the discussion (pages 744–7). This showed the great variation that was found in the conditions under which wire ropes were used, and the extreme difficulty of arriving at any conclusions whatever under working conditions.

In reply to Mr. Wingfield (page 749), the author was quite prepared to accept the suggestion that the number of points of contact between the rope and pulley might only be 50 per cent.

greater with the rope of finer wires suggested in the Paper, and accordingly the pressure per wire would be two-thirds as great as in the case of the ordinary rope instead of one-quarter as suggested in the Paper. This greater pressure would accentuate the wear and give even a shorter life than was suggested by the author, and accordingly the case of the rope of fine wires would be much worse than stated. Where the author differed from Mr. Wingfield was in estimating the rate of wear which Mr. Wingfield suggested would vary as the product of the pressure per square inch and the ratio of the diameter of the *wire* to that of the pulley. If this were the case, ropes of very fine wire would last longer than those of coarse wires, which was found not to be the case. However, all these imaginary discussions leading to different assumed results showed the desirability of having some reliable experiments carried out at the earliest opportunity.

The author's suggestion that a reduction from 40 tons to 25 tons range of stress would increase the life of materials subject to repeated applications of load by 500 times was only a very rough approximation from Fig. 56 of the discussion on Messrs. Eden, Rose and Cunningham's Paper.* Even if the increase were only 40 times, as suggested by Mr. Wingfield, it showed that the life of ropes was affected very much more by abrasion than by internal stresses.

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* Proceedings, 1911, Part 4, Fig. 56 (page 906).

# RECIPROCATING STRAIGHT-BLADE SAWING-MACHINES.

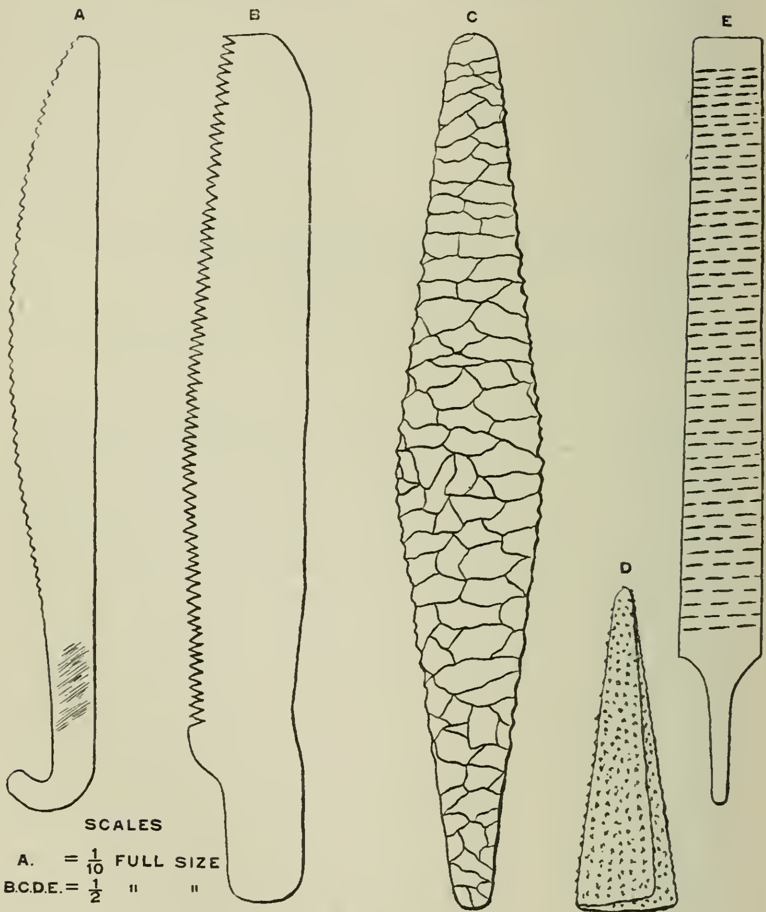
By CHARLES WICKSTEED, *Member*, of Kettering.

*Historical.*—The saw is one of the very earliest tools known to have been used, and is in fact the earliest tool that has been traced in Egyptian history. It was first found in the form of a notched bronze knife in the 3rd Dynasty or about 5,000 years B.C. (A), Fig. 1* (page 758), and was followed by larger toothed saws in the 4th to 6th Dynasties, which were used by carpenters; but there are no dated specimens until the seventh century B.C., when the Assyrians used iron saws, as shown at (B). The first knives on record were made out of flint, and were in fact saws with minute teeth (C). They must have been used for cutting up animals, as the teeth would break away even on soft wood. Rasps, which are but a form of a saw, were first made of sheets of bronze punched and coiled round, as shown in (D), but the Assyrians in the seventh century used the straight rasp made of iron exactly like the modern type (E). Coming down to modern times, the saw is possibly used more than any other tool. It has taken three

* Ency. Brit., vol. ix (11th ed.), p. 71, Figs. 17, 45, 46, and 48.

Fig. 1.—*Ancient Egyptian Tools.*

A. Notched Bronze Knife. 3rd Dynasty, 5000 B.C.
B. Assyrian, Iron. 7th Century B.C.
C. Flint Knife.
D. Rasp made of Bronze, punched and coiled round.
E. Straight Rasp, Iron, Assyrian. 7th Century.



distinct forms, both for the working of wood and metal. The straight saw, which is simply a development of the first toothed knife, the band saw, and the circular saw.

*Hack Sawing-Machines.*—The author proposes in this Paper to discuss the merits of the Straight-Blade Reciprocating Saw for metal work, and to point out that, although great pains have been taken to bring to perfection the Band and Circular type, curiously enough the Straight-Blade Reciprocating Sawing-Machine has been, comparatively speaking, neglected. The simple fact that, with a

FIG. 2.—*Star.*

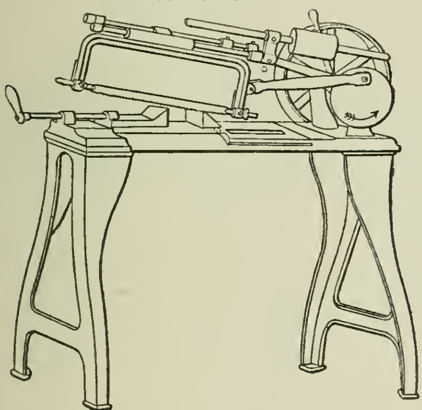
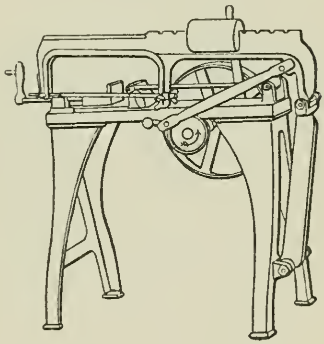


FIG. 3.—*Simplex.*



few exceptions, the cost of the Hack Sawing-Machines has in former years been only from £5 to £10 against a much higher cost for the other types, fully attests the truth of this statement. Primitive blades sold at about 3*d*. each and primitive machines sold for a few pounds were good enough for this system. The author thinks that it was Millers Fall Co., Mass., who first brought out the hack sawing-machine, which is known as the "Star," Fig. 2.

This is the crudest and lightest possible device for working a saw backwards and forwards by power instead of by hand. It is not curious that a start should be made in this way, but the curious part is that this machine, in all its essentials and with all its faults, prevails up to the present time. The connecting-rod was made of

wood, the old hand-saw bow has been maintained intact, and the guides were of the most elementary character.

The next Hack Sawing-Machine that was introduced was, the author believes, the style known as the "Simplex," Fig. 3. This is more elaborate and heavier, the old hack saw-frame giving way to something stronger. It was brought out by Mr. Hoefer and proved very successful. These two machines were the forerunners of all the present hack sawing-machines.

Another very early pattern of hack sawing-machine was the "Eureka," Fig. 4, which was made by Messrs. G. Thompson, Son

Fig. 4.—*Eureka.*

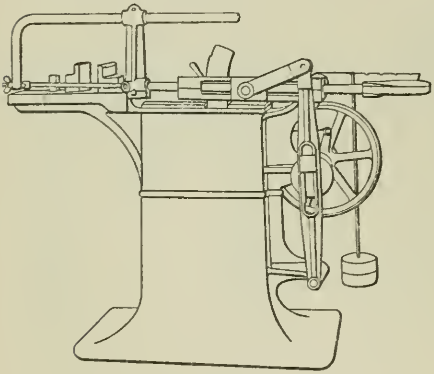
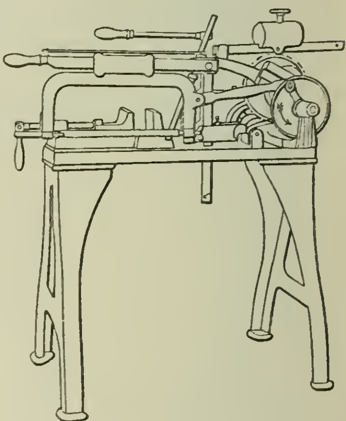


Fig. 5.—*Milford.*



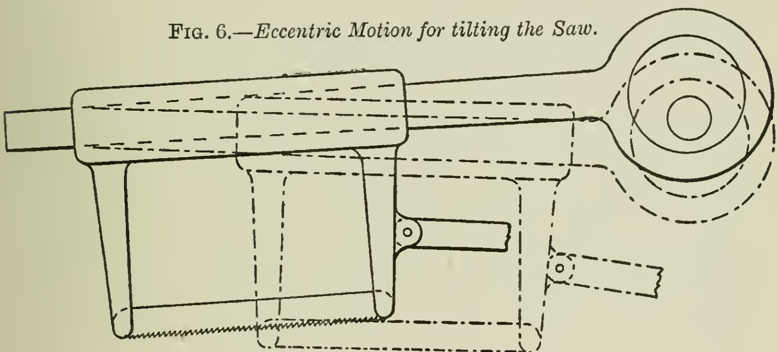
and Co., of New Haven, Conn. This is a far more mechanical and carefully designed structure; it has a solid base, and a real attempt was made to guide the saw straight, the thrust is in direct line and the upper guide being made to extend, so as to admit the large variation in the length of the blade being used. This was a magazine saw, a long blade like the band saw being coiled up and brought out for use as the working part became worn up. The coil contained 25 feet.

One of the latest machines brought out is the "Milford," Fig. 5, which is a carefully designed tool, and is in great demand.

It is fitted with a quick return and a clutch device for lifting the blade free of the work on the idle or return stroke. It has a geared drive of 4 to 1 and an automatic stop. In the author's opinion, these four designs fairly illustrate the progress of the original type of hack sawing-machine up to a recent date, although many others might be mentioned, for instance, the "Racine," which is a compact and strong tool, and the "Marvel," a strong, complicated machine with a positive feed.

Messrs. E. G. Herbert, Ltd., of Manchester, were one of the first firms who seriously took this matter in hand, with the view of making altogether stronger machines to do the work quicker and to take

Fig. 6.—*Eccentric Motion for tilting the Saw.*



larger sections. They made saws of a capacity hitherto commercially unknown, the largest being capable of taking 18 by 30 inches. They also introduced amongst other things an automatic feed for the work, but the most important feature of their machines was the eccentric motion given to the fulcrum of the saw frame, shown in Fig. 6, where, every twenty strokes or so, the eccentric on which the fulcrum of the frame is pivoted moved round slightly, thus putting the saw at a different angle to its work and bringing it on a very obtuse corner. By this means it was always working on a comparatively small surface, and it thus became much easier to saw heavy sections. The author believes that Messrs. Herbert were able to cut bars much quicker than formerly. For instance, a 4-inch bar could be cut in 20 minutes, compared with an hour taken by most

previous machines. The only disadvantage in this method is, in the author's opinion, that when first the eccentric is moved the effect on the saw is rather rough, but as quick work is often of greater importance than too great economy in saw-blades, the merits of the machine were rapidly appreciated. The author thinks that this firm was the first to use a stronger and better blade, and they were backed up by The Sterling Co., who used tungsten-steel.

Messrs. Herbert have recently constructed a tool in which the blade is set at an angle to the guides. This causes further pressure on the blade while cutting, and takes the place of the weight to that extent.

Messrs. Holroyd and Co., of Milnrow, also introduced an excellent machine for rapid work with the same object in view as Messrs. Herbert had, the difference being that in the case of Messrs. Holroyd's machine the bar turned round a little every several strokes, thus representing a small surface to the saw, as Messrs. Herbert's did, with their eccentric motion. The advantage of Messrs. Holroyd's system was that, since the bar was continually turning round, it was difficult for the saw to run, and the work would be approximately as true as that produced in a cutting-off lathe.

*Advantages.*—(1) The comparatively low cost of the machine and blade, and the fact that the blade can be made any temper to suit the work.

(2) In comparison with the Circular Saw, it will cut any depth that the frame, which holds it, will admit of. Extra depth does not necessitate extra cost of blades, and it will cut any length within 6 inches of the length of the blade. A circular saw, taking the boss in consideration, will not make a cut much deeper than one-third of its diameter, and for every extra inch in depth the saw must be increased 2 and 3 inches in diameter. It is a most expensive and cumbrous tool, necessarily fairly thick and exceedingly difficult, if not impossible, to get quite hard, and if made quite hard is of course liable to break up.

(3) In many cases the Band Saw must be cut in order to be threaded through the work and, like the Circular Saw, is almost impossible to get quite hard; moreover, it is dangerous to use if it is hard.

(4) Another advantage of the straight blade over the circular saw or a lathe cutting-off machine, is the narrowness of the cut, say $\frac{1}{16}$ inch instead of $\frac{1}{4}$ inch. This, so far as the circular saw is concerned, at any rate reduces the power taken in exact proportion to the width to be cut, and in both cases it usually saves material enough to pay for the whole operation. That is to say, if the material saved by the narrow cut as against the wide one is taken into account at the end of the day, sufficient material will have been saved to pay for the whole cost of cutting, including establishment expenses.

(5) The power taken is about one-fourth of that taken by a circular saw. One unit will cut 80 superficial square inches, which is equivalent to eleven bars of 3 inches diameter or three heavy section girders, 20 inches by $7\frac{1}{2}$ inches.

When once convinced that the straight-blade reciprocating machine had great theoretical advantages over its competitors—the Circular and the Band Saw—it did not take very long to discover the principles on which it must be made:—

(1) The blade must be kept absolutely firm and perfectly square with the work.

(2) It must be strong enough to stand all the weight that the teeth will take without breaking.

(3) The blade must be made of the highest possible quality of steel with the best cutting edge that is practicable.

(4) The machine must be well designed and work the blade without spring or vibration.

(5) Since the pressure on the blade must be considerable, an absolutely reliable release on the return stroke must be provided. In connection with this, the author has found from experiments that, unless the weight was heavy, it made little or no difference whether the blade was released on the return stroke or not, but with a very heavy weight the blade would be quickly destroyed.

Taking the points just mentioned *seriatim*, the blades have not been brought to the high quality now described without a great deal of pains and trouble on the part of the makers, who have been kind enough to assist the author in this respect. The points which appear to be fairly established are :—

(1) For ordinary work the coarse-pitch tooth, not less than 10 to the inch, is the best. They cut better, they clear themselves better, and there is better opportunity to give side clearance, which is specially necessary in the deep blades necessary for heavy machines.

(2) To make the blade strong enough to take the weight that the teeth will stand, it is not necessary to do any special tempering for this purpose. If the temper is right for the teeth, it is right for the back of the blade.

(3) Extra strength must be obtained, not by extra thickness but by extra depth. Extra thickness does not help in any way. If the blade is 20 per cent. extra thick, it requires exactly 20 per cent. more weight put on to get through the work in the same time. Theoretically, the thinner the blade the better, but in practice the deep blades must be made thicker for convenience of manufacture, because makers find it too difficult to harden deep thin blades absolutely straight, and it is evident that the deeper the blade the more difficult it is to keep the clearance. The blades used by the author vary from $\frac{3}{4}$ inch to 2 inches in depth and from 19 to 16 wire-gauge thick.

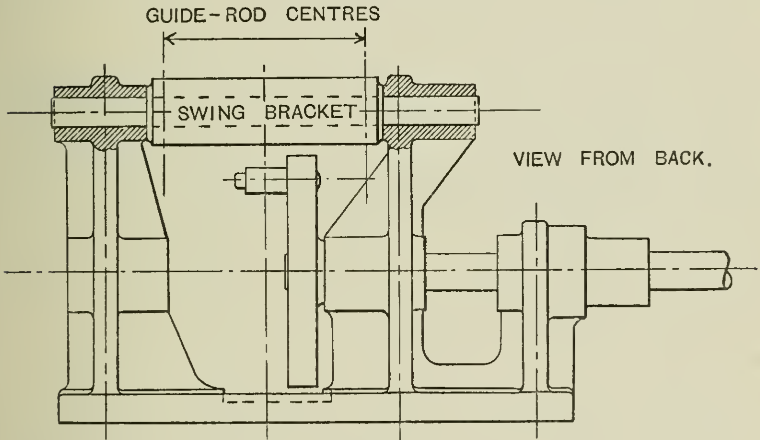
(4) The greatest weight that a tooth will take without injury must be ascertained, and the blade must then be made strong enough to take it. This weight the author finds at present to be about 7 lb. per tooth or 70 lb. per inch. A weight of 210 lb. is therefore put on a 6-inch machine, which enables it to use practically the full capacity of the blade up to a 4-inch round bar.

As the machine gets larger the proportion of weight is increased. Thus in a 15-inch machine 700 lb. is put on, so that the machine will use the full capacity of the blade when sawing a 10-inch surface. The proportion is increased in this way because it is presumed that the 6-inch machine, for instance, will principally

be doing smaller work, and that the larger machines are intended for large work which it is important to get through as quickly as possible.

*Design of Machines.*—After having dealt so fully with the blades, it is hardly necessary to point out that a machine very different from those usually employed must be designed to work them. Most of the old designs were more useful as warnings than examples. In getting out a new design, the author's ambition was to make a Reciprocating Sawing-Machine in the form of a first-

Fig. 7.—*Frame Bearing.*

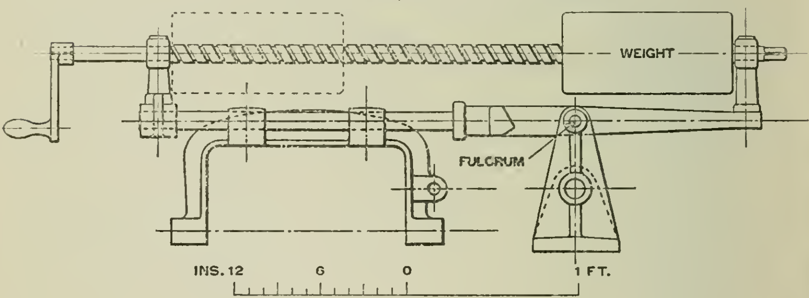



class machine-tool, on simple and sound mechanical principles that would utilize all the duty that a high quality straight blade was capable of taking. One of the greatest faults of the machines so far in use is that, following the example of the first machine that was made, the guide-frame is almost universally pivoted on the crankshaft, generally by a narrow bearing, thus ensuring liberty to begin with, which daily increases with wear. In this way was the first essential of a good machine missing at the outset. With a loose guide the saw would run, make bad work, and break the blades. To avoid this fundamental defect, the guide-frame is pivoted on perfectly independent bearings, substantial and wide

apart. These bearings have no other work than to guide the frame, and there is practically no wear whatever, Fig. 7. It was found necessary to stiffen the machine in every direction. The bed was made much wider, a bearing being thus given for the bar on both sides of the saw, the guide-bars were placed wide apart, all the bearings brass-bushed and ample, and in the case of the larger machines a vice was provided on both sides of the blade.

Weights were increased until a 6-inch machine weighed $5\frac{1}{2}$ cwt. and a 15-inch machine 25 cwt. These weights were found necessary to make the machine perfectly firm and free from all vibration. Since the weight used on the blade was so heavy, it became

Fig. 8.—*Movable Weight to prevent Vibration.*



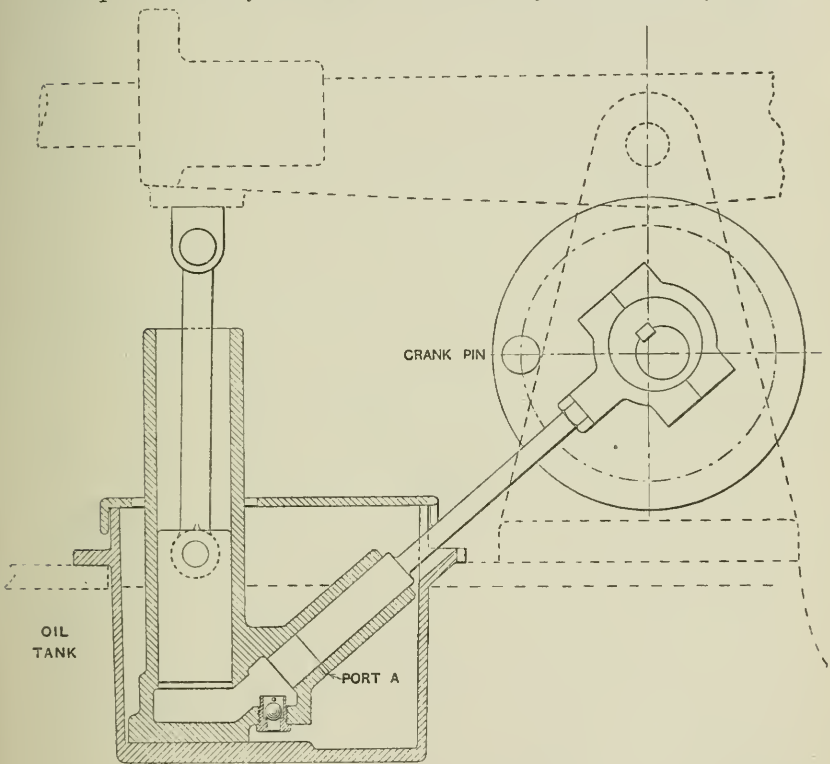

necessary to adopt a convenient method of applying this from zero upwards. This is done by sliding a weight on a bar which runs from the extreme end of the frame to a point well behind the fulcrum of the swing bracket, in which position it balances the weight of the guide-bars and frame. In the heavy machines this weight is adjusted by a quick pitch-screw, Fig. 8.

Having such a heavy weight to deal with, it was next necessary to prevent the breakage of the machine in case the blade should break, and so long as the usual small boy was to manipulate this machine it also became necessary to lift the weight by power. A perfectly reliable release was also required on the return stroke of the blade. This latter is a difficult or impossible thing to provide for satisfactorily by purely mechanical means,

as the plane of the saw varies at every stroke, so that a rack-and-catch arrangement is unsuitable. The clutch principle used on some machines is also unsatisfactory, as it constantly requires a nice adjustment which is quite beyond the capacity of the boy in

Fig. 9.

*Pump and Ram to lift and lower the Saw accurately at the extremes of stroke.*




attendance. Experiments have shown that it was essential that the saw should be lifted off its work and put down again accurately at the extremes of the stroke. If this be not done, either the output is seriously interfered with or the blade is injured. Ultimately to deal with all these points, a 4-function hydraulic ram (to be described later) was used on all the larger

machines. This was first brought out in the form of the simple ram, as shown in Fig. 9 (page 767). Here an eccentric set in time with the crankpin on the crankshaft works a little plunger in connection with the dash-pot. This plunger comes down and closes the little port A exactly at the end of the stroke, thus gently lifting the frame sufficiently off its work on the return stroke and letting it gently down again, and, as soon as the working stroke begins, leaving the full weight on the blade. There are no complications and no wear in this device, as both pistons are simply made a good fit and worked in oil. A foot-valve is provided to let the oil in when the frame is lifted by hand. It was found to perform perfectly the function it was designed for.

Modifications very quickly followed to make it useful for other purposes. The port was made small enough to convert the largest cylinder into a dash-pot and thus make it impossible for the frame to fall, as it could only be lowered as fast as the oil could be pressed through the small hole, about $\frac{1}{32}$ inch diameter in the case of the smaller machines. Soon after this, the ram was made to perform four functions by the introduction of a 4-way cock.

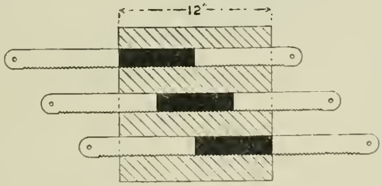
It is difficult to follow the exact action of this ram, but it will perhaps be sufficient to point out that this pump is provided with a 4-way cock with ports so arranged that, when the handle is in the horizontal or working position, it relieves the blades on the return stroke. When upright, it lifts the whole frame right off the work to any required height, the relief-port being provided to prevent it lifting too far. When in the third position, it holds the frame in the position in which it happens to be. When in the fourth, it lets the frame gently down whether the machine is standing or not. This last position is useful, as in its simple form this ram can only let the frame down when the port is uncovered, which it seldom is when the machine is standing.

The stroke in the author's machines is from 5 to 8 inches. The longer stroke adopted in the larger machines was not found necessary to get rid of the swarth, but simply because it was found advisable to reduce the strokes per minute, as the lengths of the frames increased, the momentum given by the small relief-lift becoming too much.

The question of the relative length of the blade to the stroke and diameter of the bar is interesting. A long stroke should be avoided, as it entails a correspondingly long saw blade as well as a more cumbrous machine. If there is no lift on the return stroke, the stroke must be as long as the section cut, in order to get rid of the swarth. Fig. 10 shows a 12-inch section being cut with a machine having only a 6-inch stroke. The middle part of the blade (shown black) has no opportunity of getting rid of its swarth, and will therefore take the greatest part of it backwards and forwards. With a sufficient lift on the return stroke, however, the swarth is dropped and raked 6 inches forward on the cutting stroke, and it is thus only necessary to make the teeth deep and large enough to hold the swarth created in two strokes.

Fig. 10.

*Three positions of a Saw with 6-in. stroke cutting through a 12-in. block.*
The black portion is that which always remains in the saw cut.

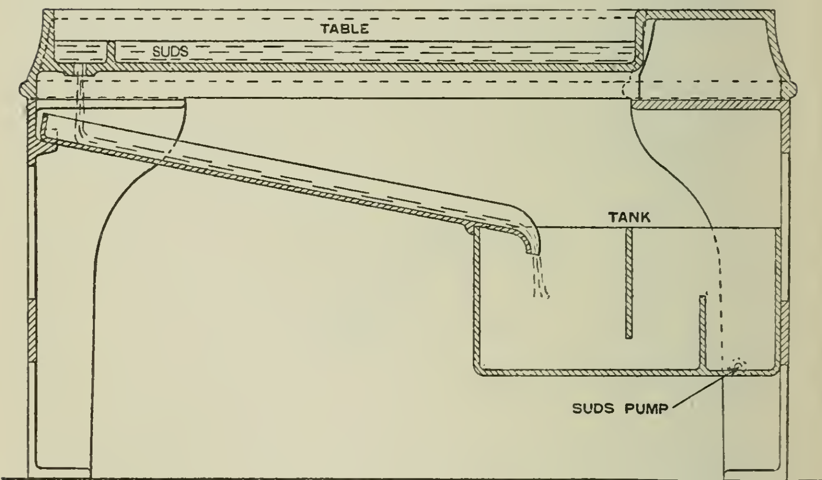



The lubricant used for saw-blades is soap-suds, and great attention has been paid to the pump and tank connected therewith. The fact that the swarth made by the saws is exceedingly fine, and that it is very much more difficult to make a pump to wear well with the suds than it is with oil, necessitates very careful provision to keep the swarth away from the pump and to make the pump as durable and as easily to repair as possible. Fig. 11 shows the arrangement of tank and connections. It will be seen that the suds are first collected by a recess in the bed, and are drained at the front end which is farthest away from the swarth, which is carried back by the blade which works on the return stroke. The suds are then conveyed through an open trough to a tank placed at the back end of the machine. This tank is divided by two weirs

(if they may be so called) into three compartments. The first weir comes to the top of the tank, leaving a space of about 1 inch at the bottom for the suds to pass through into the next chamber. This is done to prevent very light swarth on the top of the suds being washed over. The suds then go over the top of the second weir into the pump-chamber. In this way it is seen that the swarth has four distinct opportunities of being separated from the suds, and that it is almost impossible for any grit finally to enter the pump.

FIG. 11.

*Pump and Tank below machine.*

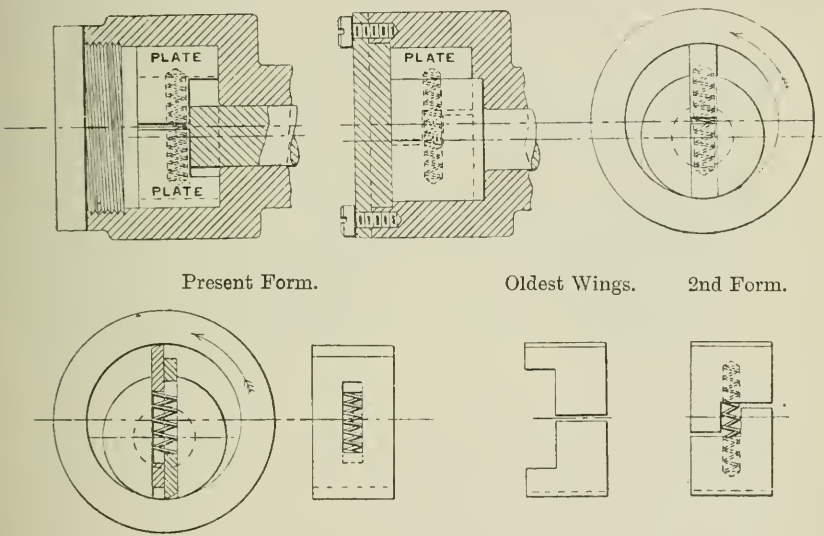



*Pumps.*—The first thing that appeared important in connection with the pump was that it must be placed under the level of the water to avoid constant priming. Wing pumps were adopted as being the most satisfactory, but the best the author could obtain were of such a construction that they would not wear well or long with suds. The simple pump ultimately adopted is shown in Fig. 12. In this pump a by-wash is provided, not with a separate valve, but simply by making the wings of the pump taper against the pressure so that, when a full discharge is not

required, the pressure of the suds will press these little wings back. The cover, instead of being screwed in, as is often the case, is simply fastened on with screws so that the barrel can be re-bored. But perhaps the most important feature of this design is that both wings go right through the head of the spindle, and thus get an ample bearing. In these wings there are two little slots of such a length that one spring put in the middle will press up the wings

FIG. 12.—*Suds Pump.*

Old Pattern, with Screw and Plate. New Pattern, easily re-bored. Old Pattern Wings.




Present Form. Oldest Wings. 2nd Form.

on each side. The advantage of this is obvious as compared with the old method where the wings met in the middle, thus leaving but a very little bearing. The gradual wearing of this bearing caused friction and the destruction of the pump. The spindle of the author's pumps is case-hardened. The new pattern of pump has been found to be very efficient.

*Results.*—The result of all these improvements is that sawing can be done practically true, say to a hundredth part of an inch in

a 6-inch bar, and mild steel can be cut at a speed, roughly speaking, varying from 1 inch to 2 inches square per minute. The breakage of the blade is exceedingly rare, and those of the best quality will often last several days.

The machines are made in several forms. The Single Sawing-Machine shown in Fig. 13, Plate 40, is constructed for sawing pieces of large dimensions quickly and correctly. Fig. 14, Plate 40, shows another variety, a machine for cutting tramway rails in position. This machine rests upon two angle-irons, which in their turn rest upon the two rails. It is only necessary to disturb the pavement for the

*Multiple Saw.*

FIG. 16.—*For cutting blanks from $\frac{3}{4}$ in. thick up to the capacity of the machine.*

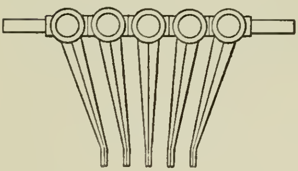
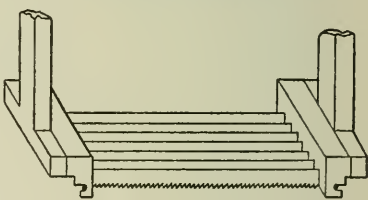


FIG. 18.—*Rack Arrangement for cutting blanks from $\frac{1}{16}$ in. to $\frac{3}{4}$ in. thick.*



frame. The main features of the machine are standard. It makes a straight clean cut through the rails, no matter how hard, in about twenty minutes, and one blade will make from about twelve to twenty cuts before it is worn out. The Multiple Saw, Fig. 15, Plate 40, and Fig. 16, is specially designed to cut off blanks for dies and similar duplicate work. By a rack arrangement, Fig. 17, Plate 41, and Fig. 18, these blanks can be cut as thin as $\frac{1}{16}$ inch; with the more usual arrangement of adjustable frames, shown in Fig. 15, Plate 40, and Fig. 16, they can be cut from $\frac{3}{4}$ inch thick to the capacity of the machine.

Another variety is called the Shaping Machine, Fig. 19, Plate 41, and Fig. 20; and this machine is provided with a table like a shaping machine on which the work is placed. By this

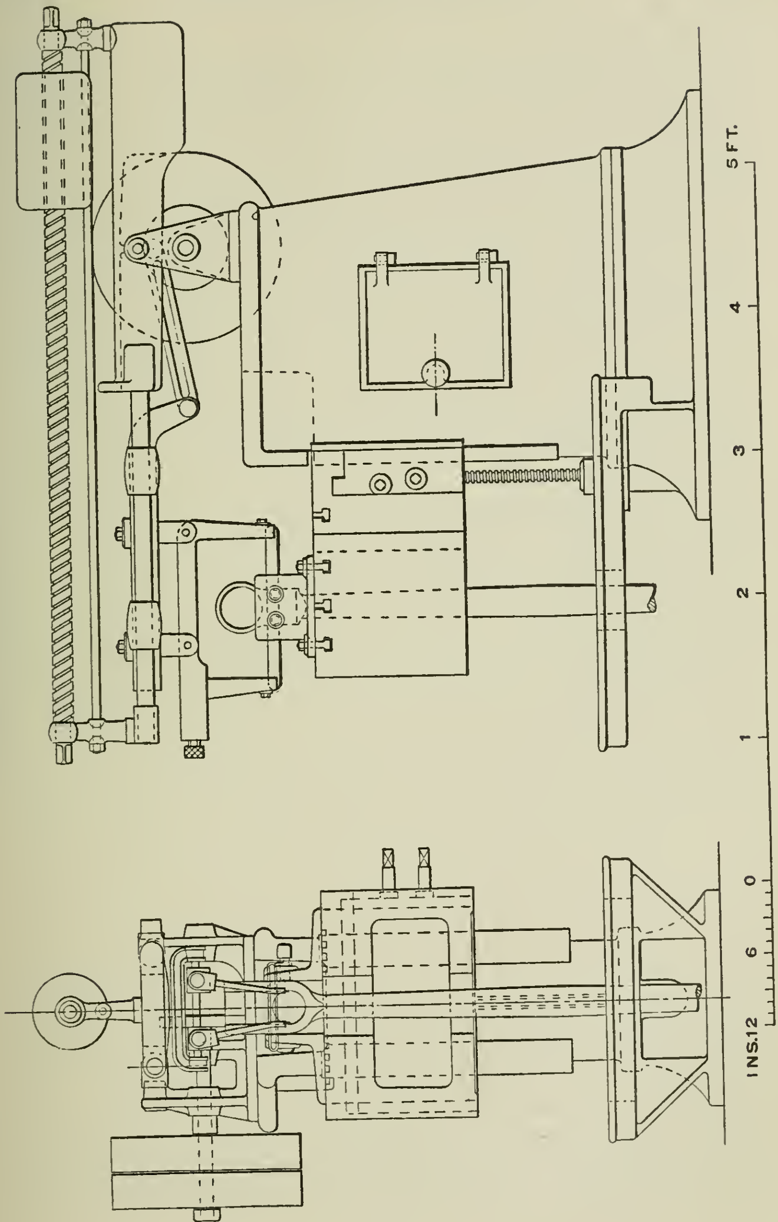



FIG. 20.—*Shaping Machine Saw.*

means the work can be brought to the required position under the blades. Two blades are provided, which can conveniently be adjusted in alignment from $\frac{1}{2}$ inch apart to the full capacity of the machine, usually about 5 inches. It is more particularly useful for cutting out joints of all sorts, splitting brasses, and so on.

Another variation of pattern is the Runner Saw, Fig. 21, Plate 41. This is specially designed to cut runners off steel castings, which are bolted on the front or the side of the table. The whole head is made to traverse a foot sidewards so as to reach the runners; it can also be lifted and put forward on the bed if necessary. The saw is brought on one side of the guides so that it will cut runners quite flush to the casting where the saw-holder does not foul. In the few cases where it would foul, it would cut flush within $\frac{3}{16}$ inch. The hard blades that can be worked with this machine make this method compare favourably in many ways with the Circular or Band Sawing-Machines.

Another variety which is in course of construction is a machine for sawing out webs of crankshafts. Here, as in the shaping machine, there are two blades, but the table is stationary, the crankshaft being bolted down on blocks to the correct position under the blades. A horizontal blade is also provided for cutting out the bottom of the crankshaft. This third horizontal blade works independently of the two vertical ones; the frame of this is constructed to hold a square file as well as the saw blade. The operation is as follows: the crankshaft must first have one hole drilled in it, say $1\frac{1}{4}$ inch or $1\frac{1}{2}$ inch diameter. It is then bolted down on the blocks in position underneath the two vertical blades, and these blades are set to work. The file is threaded through the hole and attached to the horizontal frame and the hole filed out approximately square at the bottom. To accomplish this, the lower frame is attached to a table which can be raised and lowered by hand. There is also a hand-traverse in the horizontal direction. The table is provided with a counterweight so that the filing out of the hole can easily be manipulated by the two hand-traverses mentioned. After the hole is filed sufficiently square, an operation of a few minutes, the file is withdrawn and

the blade threaded through instead. The horizontal blade is then put automatically to work. It is provided with a 4-function automatic ram, like the other, and works independently and precisely on the same system. There is usually so much less to saw horizontally than there is to saw vertically that the horizontal work will probably be finished some time before the vertical. When this is the case, the horizontal blade is removed and the frame drawn out of the way of the vertical saw before they approach near enough to foul. It will be thus seen that in the manipulation of the horizontal blade there is no time lost. A 9-inch by 9-inch crankshaft web may thus be sawn in 60 minutes to an accuracy of $\frac{1}{64}$, using about 2 h.p. The outside of the webs can also be sawn at another operation, thus saving much valuable time afterwards on an expensive lathe.

In conclusion, the author would like to state that he has simply been trying to trace the development of a most useful tool so far as it has gone. He is not of opinion that anything like finality has yet been approached, but he believes that in rapidity of work, in the endurance of the blades, and in the size of the machines, the Reciprocating Straight-Blade Sawing-Machine is still in its infancy.

The Paper is illustrated by Plates 40 and 41 and 15 Figs. in the letterpress.

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*Discussion.*

The President, in asking the members to accord a hearty vote of thanks to the author for his valuable and instructive Paper, said that Mr. Charles Wicksteed was a master in the art of designing engineers' tools, and he was sure the description given in the Paper of the work that had recently been done in that direction would be received with great interest, and would lead to a good discussion.

The vote of thanks was carried by acclamation.

Mr. JOHN A. McLAREN, in opening the discussion, said that the firm with which he was connected had used one of the author's shaping-machine saws, and as compared with the band-saw they found it had many advantages, the chief one, of course, being the two saw-blades. One never knew whether the band-saw was doing its utmost; it generally seemed to be going more or less slowly. If the man was asked whether he could not put a faster speed on, he replied that he could, but that it would knock the saw up, and in the case of a large one it meant half an hour to change it. On the other hand, if one of the author's little short saws broke, it could be changed in a few minutes, and it cost less to put in a new one than to change and sharpen the saw in the case of the band-saw.

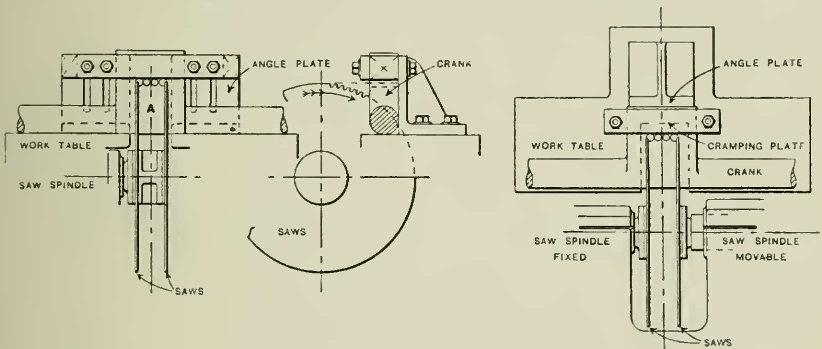
The author stated in the Paper that he was bringing out a machine to saw out the webs of crankshafts. His (the speaker's) firm did that at the present time quite successfully with their double machine. They drilled two holes along the top of the crank-pin, or sometimes one, and sawed into it. He thought that was perhaps an advantage the author's new machine did not possess, because it was very important that the metal piece that was sawed out was properly fastened. When the saw got down close to the bottom the metal began to move, and as a result the saw was frequently broken. If a hole or two holes were drilled, it was a comparatively easy matter to secure the piece of metal by means of plates and bolts. His firm found that in sawing short jobs, such as $1\frac{1}{2}$-inch bars, the saw lasted quite a long time. They had had cases of saw-blades lasting a week if they were kept on little jobs; but if the saw was put on a big job, such as a 6 or 7-inch bar it would not last nearly so long. His firm used to have a night shift always going on a large band-saw, but when the author's little saw was obtained the night shift was stopped, and the day man looked after both saws, which could easily cope with the work. The author's tool was good and cheap, and as a result of experience he had nothing but praise for it.

Mr. JOSEPH HILL desired to express his thanks to the author for bringing forward a Paper dealing with a subject which was very

important in the present labour-saving age. His own connection with cold-sawing dated back to the year 1885, so that he had had twenty-seven years' experience of it. He wished specially to call attention to the fact that there was an idle stroke in the machine described by the author. There was the lifting of the saw in the back stroke, and if it was not for the back stroke there would be no need for the lifting of the saw. He desired to emphasize that point, because the author deliberately dwelt upon the question of the circular saw and the band-saw (pages 762–3). Personally, he was present to hold a brief for the circular saw, having been the

Fig. 22.—*Sawing of Crank-webs.*

*Crank Vertical, Elevation.* *Crank Horizontal, Plan.*




inventor of a flush side-saw which he brought out in 1885, and he had thus had a very large experience of such blades.

It seemed to him that a certain amount of doubt ran throughout the Paper. There was a doubt, as the members had already heard from the author, of the saw being able to stand more than a week's work. He could not help thinking that the author would have been wise if he had not mentioned so much about the circular saw. There were many thousands of these saws at work, and although he might be accused of egotism he would like to say, incidentally, that he had been paid no less than £6,500 in three years for circular saw-blades. With regard to the author's idle saw-blades, he remembered that a Scotsman once said to him when speaking of

(Mr. Joseph Hill.)

the slotting machine, "Hang the up-stroke!"; and personally he now said, "Hang the back stroke!" In his opinion there were things in the author's machine that were not wanted because of the back stroke, and he certainly thought that a circular saw if it was properly used, properly sharpened, and properly cared for, was one of the best tools that an engineer could possess.

Referring to the remark made by Mr. McLaren (page 776), he thought it might be of interest if he stated that the sawing of crank-webs was a very easy matter. It was possible to have two saws on one spindle, the crank being held in the following way:—The piece was shown at A in the diagram, Fig. 22, and the grooves were cut right through, so that the crank was readily cut. There

Fig. 23.

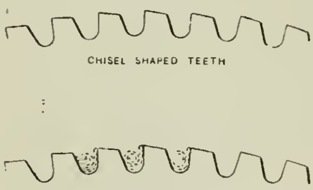
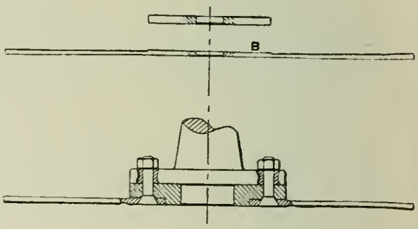



Fig. 24.




was a large firm in Birmingham which was sawing crank-webs 10 inches thick, two cuts at a time, with one of his machines, and taking only one hour to do the work. In this instance he used a shape of tooth, shown in Fig. 23, that was not of the ordinary kind. The tooth was chisel-shaped, and the reason it was made like that was for the purpose of obviating the cuttings clogging in the teeth of the saw, because the cuttings would curl round as shown, and by so doing force themselves out. In the machine that was exhibited, the author had carefully withheld how he went on with his saws when they got clogged up with soft material. In a circular saw no hydraulic apparatus was required, neither was there any lift, as there was no back stroke. It must be remembered also that in the author's saw, or in a band-saw, they were cutting with a ribbon of steel in tension, and the tension it would stand depended entirely upon the toughness or temper of the saw-blade. If the blade were

too hard it would snap and break, but if it were properly tempered, it was possible to do fairly good work with it. A circular saw could be pushed against the work, and at the same time everything depended on the quality of the blade and how it was tempered to stand the pressure put upon it. The resistance or strength of the saw-blade to withstand the pressure exerted in feeding the saw into the cut was enhanced by the way the saw was held on the spindle.

A flush side-saw was not limited only to attaining simply the flush side. The main thing was to retain the saw in a straight cutting-line so that there was no friction on either side of the blade when in the cut; thus it was absolutely certain to make a true cut. As the author mentioned, if one desired to strengthen the saw, it must have it in depth, but not in thickness. It was most necessary to have thickness in order to get through a solid piece, and he had no doubt the author found that out in cutting through solids; therefore there must be good clearance on the blade. He desired to show the best method of holding a circular saw, which could not be done with a ribbon saw. Fig. 24 showed the blade which was ground so that a clearance in the saw was obtained.

The hardening and hammering of a saw-blade was a delicate matter, no matter what class of steel the blade was made of; it had its own way of buckling, whether it was done by passing into the hardening bath obliquely, or whether it was put in flat, or whether the oil in the bath was of a varying temperature. The cooling had also to be borne in mind. The hammering of the blade was a skilful business. When the hammer-smith took the saw to straighten, he was invariably compelled to put on the portion that was intended to be held by the washer a lot of peck-hammer marks for the purpose of drawing the saw, so as to make it perfectly true. This process of hammering was done at intervals during the hollow-grinding process; therefore, as far as the saw-smith was concerned, when it came from his bench it was completed and considered true. But if the blade-saw was put in between two true washers and screwed up, the tendency was for it to buckle, therefore no straight cut could be made, no matter how true the

(Mr. Joseph Hill.)

saw was considered to be. Formerly he used to have his saws ground with a flat space on the side, B, Fig. 24, which he called the left of the saw, but, all the way to the centre on the right side, holes for rivets were made in the saw whilst the blade was in the soft. Having prepared a steel washer true on both sides, he took and bolted this washer to the saw-blade, it having corresponding $\frac{1}{2}$-inch diameter holes, and he bolted the washer temporarily to the blade of the saw. When he had bolted it up, he found out how much the saw had buckled and how much the saw would give way to the plate. In many cases he had had a saw dished as much as $\frac{3}{4}$ inch in a concave way, or it might sometimes be dished in a convex way. To correct this buckling, the method he adopted was as follows: He bolted a washer to the saw-blade, but supposing, after he had bolted the saw to a plate, he found that the saw was hollow, then his method came in. He inserted, between the saw and the plate, a piece of a very hard specially prepared brown paper, in such a way that he could regulate the buckle by packing between the washer and the saw. Sometimes it took two or three thicknesses of paper to make the saw perfectly true on either side, the washer being permanently secured by rivets in place of the temporary bolts, so that when completed he had practically ensured the saw being theoretically true on the spindle before it was fixed on the saw spindle, and before it went away from the fitter's bench. He quite understood that the saw would have to be perfectly clear, or it could not cut straight and true.

The points he had referred to were very important. Personally, he gave his saw the name of the flush-side circular saw, while the author's was a straight-blade saw. With regard to the author's machine, as the feed of cut was given by the weight, and the weight had been taken off the saw-blade only in the return stroke, what happened to the cutting stroke when the blade was broken? If it was going forward, the pump had not commenced to work and, therefore, it must fall. He was sure the return of the saw in the back stroke, or easing the pressure given upon the saw by the weight, was the invention of one clever appliance to obviate the great disadvantage of the back stroke. If it were possible to do

away with the back stroke, the machine would be a far better one. As the author had dwelt so deliberately upon the question of the circular saw, he (Mr. Hill) felt it was his duty to speak in its defence.

Mr. Louis W. Smith said that having during the last few years had occasion to give special attention to the quickest and most economical method of sawing, both on band-saws and hack sawing-machines, he thought it might be of interest if he gave a few figures of the comparative cost of running, not only taking labour into account but all establishment charges also. It seemed to him that, although Mr. Hill naturally thought a great deal more of the circular saw than any other make of saw, there was work more suitable for the hack-saw than for the circular saw, and now that during the last year or two the author, amongst others, had given them such an efficient tool for using the hack-saw blade, he could not but think there was a great future for this machine tool; and he would be sorry to hear it termed an idle machine when it was so busily engaged on useful work in so many engineering shops at the present time.

He had worked out during the last few months a comparative statement of the cost of running a band-saw or a group of band-saws and hack-saws of the modern type as represented by the author's design, and he found that, disregarding the slight difference in amount of power required for driving the machine, which was not very material but including, say, $7\frac{1}{2}$ per cent. depreciation, 5 per cent. interest on capital, wages, saw-blades used, and lubricant required, the material sawn being similar for both types of machines, say, 0·25 per cent. carbon, the hack-sawing machine could be run for two-thirds the cost of a similar sized band-saw machine; in the speaker's case £90 total for one band-saw as against £60 for the hack-sawing machine, and this notwithstanding the cost of the hack-saw blades themselves being considerably more than the band-saw of the other machine. If, as he hoped, a better quality of hack-saw blades would shortly be obtainable—blades which were more carefully tempered, and perhaps also later on

(Mr. Louis W. Smith.)
there would be the possibility of having them recut, the hack sawing-machine would be then still more economical in use. In his opinion, there was scope for great development in the author's machine, especially in the cutting out of crank-shaft webs, and he sincerely thanked the author for having produced a machine which did not break so many saws, and was built on modern machine-tool lines.

Mr. CHARLES WICKSTEED, in reply to the many complimentary remarks which Mr. John A. McLaren had made (page 776), said that if the block he proposed to cut out of the web gave trouble when the saws went through the bottom, there would be no difficulty in dealing with the matter in one way or other. Perhaps the best plan would be to leave about $\frac{1}{16}$ uncut, so that the block would not fall out; this uncut piece would only be left on one side, as the other side would go through into the hole. There would, therefore, be no difficulty whatever in afterwards taking the block out.

Owing to the short time at his disposal he could not follow Mr. Joseph Hill into all the points he made (page 776), but was surprised to hear Mr. Hill boast of having sold £6,500 worth of circular blades in so short a time, as the only inference that could be drawn from this large sale was that the working of his machines was very expensive to the users. As far as the idle stroke was concerned, it was a good thing for the makers of circular saws that there was an idle stroke in the reciprocating saw, otherwise they would have no chance. Every one knew that the advantage of a circular saw or a band saw was that it was continuously cutting, and that one of the disadvantages of the reciprocating saw was that it was not. Mr. Hill would have found out what happened when the saw-blade broke on the cutting stroke, had he given more attention to the subject of the Paper and less to the circular saw. The contingency of the blade breaking on the cutting stroke, and consequent danger of the breakage of the machine by the sudden fall of the frame with a heavy weight on it, were provided for by

a dash-pot, which was a part function of the oil ram, and explained in the second paragraph of page 768.

He was very much interested in what Mr. Hill said about his circular blade and the teeth that he designed to prevent them from choking; it showed that Mr. Hill had taken a great deal of pains and showed great skill in bringing out a good blade. It also made one think of the great difficulties there were in making a blade true. The fact of the matter was that both the circular and the band saw in ordinary use were liable to run very much. His silence on the question of the choking of the blade with soft material was explained by the fact that there was no choking. The idle stroke with the lift on the return, which Mr. Hill made such sport of, was one of the reasons of this; the teeth were cleared at every stroke, the lifting motion pulling the cuttings from between them. He had never experienced any trouble in that respect, no matter what class of material they were cutting.

With regard to the question of the shape of the teeth, he desired to point out that up to the present comparatively little attention had been paid to straight blades, and that was one of the reasons why they had not so far made anything approachingly as good as they probably would do before long. It must be remembered that he had only just begun the work; it had taken him all his time to get blades made of the strength he wanted, quite apart from the question of the teeth. He had personally been experimenting, so far not successfully, with the object of getting a really good milling cutting tooth, instead of the sharp cornered tooth at present in use, and he did not know whether or not he would succeed. As the subject had been raised, he would like to say that a blade exactly the same as the dished blade, that Mr. Hill described, had been made for a hack-saw already. It was expensive, costing four or five times as much as the ordinary blades, but it was wonderfully successful; it had been at work for a week, and seemed as good at the end of the week as it was at the beginning. The improvement of the blade was in the future, and if he had had the time, and anything approaching the skill that Mr. Hill had in bringing out his blades, he believed he would

(Mr. Charles Wicksteed.)

have produced a blade many times better than the best at present in use, although he feared he could never compete with Mr. Hill in the successful application of brown-paper packing.

He had been very much interested in what Mr. Louis W. Smith had said, and thanked him, and all of those present, for the kind attention they had paid to his Paper, and also thanked Mr. Hill for the good humour of his drastic criticism.

*Communication.*

Mr. WILLIAM POULTON wrote that the firm with which he was connected—the Baltic Steel Works, Sheffield—were users and also makers of saw-blades of every description, and had experimented with and perfected several kinds of tools of this description for cold-cutting. It was interesting to note the author's remarks (page 762) as to the use of tungsten-steel blades made by the Sterling Co. His own firm's most successful hack-saw blades were made from genuine high-speed steel, hardened on the teeth only by a special process and used in a Racine machine cutting sections, rounds, etc., of annealed high-speed steel. Ordinary blades of most makes were not satisfactory on this class of steel, which, owing to its density, was very severe on saws, the life of ordinary hack-saws in almost every case being very short.

With regard to the remarks on page 762 concerning the comparison of circular saws with hack saws, the author could not be aware of the improvements which had taken place in this direction. As to the difficulty mentioned in getting these blades hard in certain qualities of steel, the writer was in agreement. On their latest improved saw, made throughout of genuine high-speed steel, the blade was not hardened throughout, but only to a depth of 1 inch or 2 inches round the circumference, the rest of the blade being stiffened to withstand buckling or any other

form of violence which might arise in the life of the saw. This was not an experiment, but an actual fact, and four or five times duration of life of saws was guaranteed over any ordinary crucible carbon cold saw; with regard to hardness of teeth of the saw, this had been found to be of a hardness of 600 on the Brinell test. Over 200 of these saws had been supplied during the present year by his firm.

# COMMERCIAL UTILIZATION OF PEAT FOR POWER PURPOSES.

By H. V. PEGG, of Belfast.

The question of the utilization of Peat Fuel for power purposes has received a large amount of attention from engineers for many years past. Efforts in this direction have mostly taken the shape of some form of preparation of peat fuel in order primarily to get rid of the superabundant moisture in the fuel. Very large sums of money have been spent on peat-preparing machinery with generally very inadequate results; hence it has always appeared to the author that, in order to bring the utilization of peat to a commercial level, the first consideration would be the utilization of the peat as far as possible in the condition in which it leaves the bog-lands without any preliminary and expensive machine treatment.

The author had the opportunity about seven years ago of experimenting with air-dried hand-cut peat fired into a special form of gas-producer. With all gas-producers using bituminous fuel the main trouble is to get rid of the tarry by-product. In this instance the gas-producer was arranged to work intermittently, there being periods of "blowing" during which the fuel in the producer was urged to incandescence, and periods of gas-making during which the tarry by-products were passed through the

incandescent fuel where they were split up into gas. The chief difficulty experienced with this plant was the high thermal value of the gas generated, about 330 B.Th.U. Owing to the high and varying percentage of hydrogen in the gas, it proved unsuitable for use in the works gas-engine; and although the plant was running more or less continuously for ten days driving the whole works, very considerable trouble was experienced, not only in the engine, but also in the plant, owing to the varying moisture content of the peat, the producer plant being decidedly sensitive in regard to this latter point.

From the experience then gained it appeared evident that it would be wiser to extract the tar from the gas, rather than to try to utilize the same by converting it into gas, and further that the producer must be comparatively non-sensitive to the amount of moisture in the peat fuel. Some two years ago the author discussed the question of the utilization of air-dried peat fuel with Mr. Hamilton Robb, of Portadown, who, having large supplies of such fuel convenient to his factory at Portadown, was strongly of opinion that it should be possible to utilize such fuel in order to generate the power required in the factory. As the result of various tests run with an experimental plant at the works of Messrs. Crossley Brothers, Openshaw, a special plant was eventually manufactured by them under their designs and patents and to the author's specification. This plant, which has been running since last September, has been so often dealt with in the daily and technical press that there is no need for the author to dwell upon the details of the plant, but he proposes to make a few remarks in regard to the difficulties experienced.

Air-dried peat is not a very convenient fuel to fire into the producer, and as it was uncertain whether it would be possible to burn the fuel direct in the form in which it came from the bog-lands, provision was originally made in the plant to deal with peat fuel prepared by being reduced in size to blocks of about 5 inches cube, but it was found possible to dispense with the preliminary treatment, and the construction of the plant was thereby considerably simplified.

As regards the general running of the plant, last October it was subjected to a test run of six hours' duration with a load of 250 b.h.p., the peat consumption per b.h.p.-hour averaging 2·55 lb., the peat fuel containing 18·98 per cent. of water; this was with both producers running, although the load was considerably below the total capacity of the plant. When necessary, it has been found that the above load can be safely carried with either producer working singly, and the plant has run under these conditions for several days.

It will be noted that the percentage of moisture in the fuel during the above test was unusually low. This was owing to the unusually dry summer of 1911. During November and especially December last the fuel fed to the plant was extremely wet, as the rainfall in those months was very heavy and the fuel-supply was and is entirely exposed to the weather. The plant, however, worked just as well with sodden peat as it did with the drier peat, the only difference being the amount of fuel consumed. The amount of water in this "sodden peat" varied considerably from day to day, and the exact percentage was not arrived at; as near as could be estimated, it was at least 70 per cent.

The separation of the tar from the gas was the chief difficulty to be overcome; it was found far better to rely on an ample water-spray through which the gas passed rather than any form of a coke-scrubber, as the coke rapidly became clogged with tar. The main portion of the tar was thrown out into a tar sump by a centrifugal tar-extractor; but unless the gases were subjected to a thorough washing and cooling by the water-spray above referred to, it was found that a certain proportion of tar got past the extractor, collected in the gas-mains and finally found its way into the gas-engines. It was a matter of experiment as to the precise amount of water sprayed into the cooler which was necessary, in order to insure that the tar vapour should be sufficiently condensed before reaching the centrifugal extractor, so as to enable the extractor to effect the needful separation. As now arranged, the proportion of tar in the gas after passing the extractor is small, and the engine-valves do not want cleaning out more than once a week.

When first started, the plant generally and the producers especially required a thorough cleaning once a week; at the present date the plant can be run if necessary for three weeks without cleaning, though the weekly cleaning generally takes place as a matter of policy. This result has been obtained owing to the increased amount of washing-water used, which now amounts to about 7 gallons per b.h.p. per hour. The proportion of tar recovered is about 5 per cent. of the weight of fuel consumed, and during the initial stages of the running of the plant a certain amount of this tar was sold to tar-felt manufacturers at a price of 35*s.* per ton, but sales in this direction ceased owing to an, at present, ineradicable pyroligneous odour which persistently clings, not only to the tar itself, but to all the various oils distilled therefrom.

Experiments have also been made with the tar in oil-burning boilers, but owing to the very high percentage of water in the tar —up to 50 per cent.—and the large quantity of solid matter also present, a very large amount of preliminary treatment is necessary. For a considerable period the tar at Portadown was used mixed with coal and burnt under a Stirling boiler; the precise heating value of the tar so consumed has not however been ascertained. At the present time the whole factory at Portadown is run entirely on peat fuel, the consumption being about 44 tons per week, of which the producer plant takes about 22 tons. The nature of the peat varies considerably; with good black heavy peat the weekly consumption for all purposes drops as low as 35 tons; and with light top peat from the surface of the bog-lands the consumption rises to 54 tons. It is also interesting to note that the quality of the peat is reflected in the carrying capacity of the barges, which bring a load of 35 tons with heavy peat and 24 tons light peat. The peat is unloaded from the barges and conveyed to the producer platform and boiler-house by a transporter. Clinker troubles are not often experienced, and only when burning the inferior grade of peat, the presence of sand in the fuel causing the trouble.

The author is indebted to Mr. W. A. Mullen, manager at the factory of Messrs. Hamilton Robb, Ltd., for the following figures

in regard to the cost of fuel, these figures being given on the 12th of June last:—

| | £ | s. | d. |
|---|---|---|---|
| *Cost of running Factory on Coal per week:—* | | | |
| $8\frac{1}{2}$ tons of Anthracite @ 35*s*. . . . | 14 | 17 | 6 |
| 19 tons of Steam Coal @ 17*s*. . . . | 16 | 3 | 0 |
| | £31 | 0 | 6 |
| *Cost of running Factory on Peat per week:—* | | | |
| Say up to 50 tons of Peat @ 6*s*. . . . | 15 | 0 | 0 |
| Weekly saving . . . . | £16 | 0 | 6 |

Allowing for 15*s*. for extra labour, the net weekly saving figures out at £15 5*s*. 6*d*.

The author would here refer to the letter in *Engineering* of the 26th January last, in which the General Manager of the Power Gas Corporation, Ltd., gives some very interesting particulars in regard to peat plants, more especially an ammonia recovery plant working in the South of England. It would be of great interest if some figures as to the working costs of this plant could be laid before this Meeting. It will be noted that plant is worked with ammonia recovery which would mean a very much larger plant than that at Portadown. The amount of the nitrogen in the South of England peat is apparently high, and would appear to be considerably more than in the peat used at Portadown, analysis of which is appended hereto, together with analysis of the refuse tar and gas from the producer.

## Analysis of Tar made by Messrs. Totton and Hawthorne.

*Sample of Tar No. 2.*

The sample was grey when received, but very quickly turned black. On distillation it yielded:—

| | Per cent. |
|---|---|
| (1) Water . . . . . . . . . . | 37·2 |
| (2) Light oils (distilling below 230° C.) . . . . | 5·8 |
| (3) Middle oils (distilling at 230–270° C.) . . . . | 8·3 |
| (4) Heavy oils (distilling above 270° C.) . . . . | 23·2 |
| (5) Coke . . . . . . . . . . | 17·8 |
| (6) Loss . . . . . . . . . . | 7·7 |
| | 100·0 |

Much frothing occurred until the water was distilled off. Towards the end the temperature went higher than a mercury thermometer will record (360° C.). The different fractions obtained were as follows:—

(1) Water, faintly acid to litmus. Phenol could not be detected.

(2) Light oils (below 230° C.) became rapidly dark red in colour. Specific gravity of crude liquid 0·930.

(3) Middle oils (230–270° C.) became dark red. Specific gravity of crude liquid 0·944.

(4) Heavy oils (above 270° C.) on standing, crystals of paraffin wax separated out to the extent of 5·42 per cent. of the fraction (= 1·26 per cent. of the original tar). The specific gravity of the liquid portion of this fraction was 0·906.

## Analysis of Sample of Peat.

(Received on 14th September from Mr. Hamilton Robb, Portadown.)

*Proximate Analysis.*

| | Per cent. |
|---|---|
| Water | 18·98 |
| Volatile Matter | 55·17 |
| Fixed Carbon | 24·75 |
| Ash | 1·10 |
| | 100·00 |

*Ultimate Analysis.*

| | Per cent. |
|---|---|
| Carbon | 44·60 |
| Hydrogen | 5·42 |
| Nitrogen | 0·97 |
| Ash | 1·10 |
| Moisture | 18·98 |
| Oxygen (by difference) | 28·93 |
| | 100·00 |

Analysis of Average Sample of Gas during a Ten-Hours' Trial.

*Moisture in Fuel 26 per cent.*

| | |
|---|---|
| $CO_2$ | 10·6 |
| CO | 21·0 |
| $H_2$ | 13·0 |
| $CH_4$ | 3·7 |
| Total Combustible | 37·7 per cent. |
| Calorific Value (calculated from analysis) | 144·0 B.Th.U. |

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*Discussion.*

The President, in proposing a hearty vote of thanks to the author for his Paper, said it dealt with an extremely important subject, because it was of supreme importance to Ireland that the peat deposits should be utilized to the fullest possible extent. It appeared from the Paper that, after overcoming great difficulties, success had been achieved which might have far-reaching results in the future.

The resolution of thanks was carried by acclamation.

Mr. Bowman Malcolm, in opening the discussion, agreed with the author that the only chance of success in the use of peat in Ireland was that people should be able to use it without any treatment. Some years ago a system was inaugurated at Kellswater of drying peat for household purposes. The peat was first worked up into a pulp, then squeezed through a die, and it came out quite hard and apparently quite dry. An experiment was made with that particular peat on one of the Northern Counties Committee's locomotives, but they were able to get only 7 miles out of Belfast on the road with a tenderful, and if the road had not been downhill they would never have got back again. There was no man living who

(Mr. Bowman Malcolm.)

could shovel it into a fire-box fast enough to keep up steam. Various experiments had also been tried in the south of Ireland, and he believed the great difficulty experienced was in following the bogs with the peat-preparing plant. The adjacent turf was soon used up, and one was perpetually moving the plant to follow it up. He was unable to speak with regard to the particular plant at Portadown described by the author. It seemed to be very economical, but it must be remembered that it was simply used for the manufacture of power-gas for power purposes. Personally, he had not very much hope of the ultimate success of using peat.

Mr. William H. Patchell (Member of Council) said that the press to which Mr. Malcolm had just referred appeared from the short description to be what was known as the Extrusion press, which was used in the German lignite or brown coal industry to a very large extent. He would like to refer to the analysis given on page 793, where 26 per cent. of moisture in the peat gave 13 per cent. hydrogen in the gas. The author remarked that it made practically no difference to the operation of the plant whether the peat was cut in the dry summer or in December, but as there were 8 inches of rainfall in the latter month he presumed it must make a considerable difference in the hydrogen content of the gas. It would be of interest if the author would state what the hydrogen content was of the gas in the winter when the peat was 70 per cent. water sodden. There was a little difficulty in working gas-engines, in that it was impossible to work the same compression conditions with a high content of hydrogen gas as with a lower hydrogen content.

Mr. George Andrews said he was not a mechanical engineer, but some years ago he took great interest in the question of the utilization of peat. He remembered seeing a process, further details of which the author might perhaps give, which dealt with the peaty mass by electric treatment for eliminating what might be called the latent water. It was quite easy to get rid of the loose water by centrifugal extractors, but it was found that in the

molecules of peat there was a large amount of water which remained stubborn and would not come out under ordinary treatment. The electric process passed a current through the mass, and broke up the molecules of peat so that the water was very much more easily expelled. That process at one time seemed to present a good many possibilities, but nothing seemed to be heard of it lately. The Department of Agriculture took it up, and had a plant laid down somewhere in the midland counties of Ireland. The difficulty, however, seemed to have been the cost of the preliminary treatment, and the subsequent treatment of the peat after being partially dried by the process was to form it into briquettes for sale. It was claimed that these briquettes were of very high thermal value.

Another point that had occurred to him, during the reading of the Paper, was whether a use could not be found for the tar, which was obtained as a by-product from the peat when used for making gas, by making tar-macadam for roads. The smell of the woody acids would not matter much on Irish roads, because there were at some seasons of the year a good many other smells there already.

Mr. Walter Dixon said that the members continually heard, as they had heard that morning, of experiments which were carried on years ago, but he thought it was necessary to face modern developments in regard to the question of utilization of peat. He thought it must be taken for granted that, as an economic question, peat could be dealt with in a practical manner. The question was one of great importance to Ireland, but it was of much more importance to other countries such as Canada and Sweden, both of which had carried out very large investigations and had arrived at certain results. People on the other side of the Irish Channel did not appreciate the question of the utilization of peat, because cheap coal was available in England and Scotland; and as long as coal could be purchased there at anything like present prices the utilization of peat need not be discussed so far as those countries were concerned. The author had shown that in Ireland, where 17s. a ton had to be paid for ordinary coal and 35s.

(Mr. Walter Dixon.)
a ton for anthracite coal, peat at 6*s*. a ton could be dealt with as a commercial matter, and he was prepared to accept the author's statements on that point.

The question of the utilization of peat had been investigated on two direct lines. One line of investigation was whether the whole of the water could be taken out, the peat being so compressed or dealt with that it could be handled in a solid form in the same way as coal. That, as an experimental matter, had been carried to a quite successful issue. It was a well-known fact that it was impossible by any kind of mechanical treatment or mechanical drying to abstract the whole of the water from peat. Some of the water in peat appeared to exist in some chemical form which prevented the whole of the moisture being abstracted. Many thousands of pounds were being spent at the present time on a process which practically amounted to boiling the peat bog, that is, disintegrating the whole of the peat, drying it out in a mechanical process, and obtaining the residue. The peat was then in a pulp form, and could be manufactured into briquettes. That process was being carried out in Scotland with the idea of manufacturing peat to compete in a solid form with coal.

The Paper had brought before the members the important development that had taken place in the industry in the last two or three years. To speak of anything before the last two or three years was to be comparatively out-of-date. The author had shown that it was possible at the present time satisfactorily to utilize peat without any mechanical process in drying. Directly a mechanical process of drying was adopted, expense was incurred. Mr. Pegg had shown that in a special gas-producer plant which had been erected, peat could be utilized and gasified. One of the objects of the author in giving the Paper was to obtain further information, in order that he might go a step further and deal with the question of the recovery of by-products. Personally, he (Mr. Dixon) had given a good deal of attention, both theoretical and practical, to that subject. Quite recently his colleagues and himself had, in a practical way over an extended period, produced by-products from 1 ton of dry peat which had a marketable value of,

say, 17*s*. He quite appreciated the importance of that statement—that from 1 ton of dry peat there had been recovered in a practical way, that is, dealing with many tons of peat, sulphate of ammonia which had a market value of 17*s*. The recovery of by-products and the recovery of gas from peat, when it was in the state to which the author had referred, was a mechanically satisfactory process. The only question was: Was it a commercial one? The first trouble experienced when speaking of peat as a commercial product was that every farmer who owned 3 acres of peat bog, which he hitherto had regarded as useless, thought he was a wealthy man; so that the price of peat which, during the period of the investigation, was regarded as being almost negligible, immediately obtained a value which was altogether artificial.

The question of utilization depended, as the author knew, on so many subsidiary questions that, while it was practical to deal with it as he had done, and also to deal with it in a much larger way, the question had to be asked: Was it commercial? He thought it could be shown that, worked on a big scale in a country like Ireland, and given a sufficient quantity of peat, a sufficient demand for power, and sufficient capital, it could be made a paying proposition to transmit power from some of the peat bogs in the centre of Ireland for dealing with the Belfast power question; but it required a very large amount of capital, and it was necessary in its present state to take very large risks. One serious trouble to which reference had been made was that of the available bog retreating from the works.

The author had shown that he could deal with peat containing various quantities of water. He had shown how he could deal with peat containing 70 per cent. of water in his plant, but the size of the plant would grow very considerably when 70 per cent. of water was dealt with, and in that way the capital question came in. Unfortunately the peat was not usually near manufacturing centres, and the capital expense involved in the utilization of the peat was, he thought, one of the difficulties to be faced. One of the speakers at the Dinner on the previous evening observed that engineering was run by financiers. He thought the question of the utilization

(Mr. Walter Dixon.)

of peat was one which would also have to be run by the financiers, and that had hitherto been the case in Ireland. One or two projects had been run by the financiers, and they had come to rather a disastrous end. His own feeling was that, in view of such disasters, one was justified in asking the author whether the plant in question was still running. As a matter of fact, he (Mr. Dixon) had had an opportunity of seeing the plant, which was not only running, but was working under conditions and circumstances which showed that it must be running constantly and satisfactorily, inasmuch as some hundreds of workpeople were dependent upon the power. This latter point in itself was evidence and confirmation of what he (Mr. Dixon) had already said, that there was now no need to further ask the question as to whether peat could be satisfactorily dealt with in the manner described in the Paper, and he did not see why in Ireland there should not be a large extension in the utilization of plants of such size. As to whether they could be economically worked in smaller sizes, or much larger sizes, depended on points which would require special consideration in each particular case. In conclusion, he congratulated and thanked Mr. Pegg for the Paper.

Mr. JOSEPH HILL said he would like to know how the analysis of tar (page 791) was prepared. It would also be interesting to know whether that tar had been used as a tar for spraying on roads, and whether it was not better than the article at present used for that purpose. The author's opinion would also be of interest as to whether he considered tar prepared from peat was a vegetable compound or a mineral compound.

Mr. LACEY R. JOHNSON (Canadian Pacific Railway Co.) said unfortunately he had given very little attention to the subject of the commercial utilization of peat. During the past few years a great deal of interest had been taken in the subject, and many experiments had been made in regard to the preparation of peat from some of the boggy districts in Canada. The experiments had met with more or less success—probably less than more. The

Canadian people probably required the benefit of the experience that had been obtained in Ireland to guide them in the direction they should follow in their experiments. There were large districts in Canada, which were a long way from the coal fuel supply, but new things were being found every day. New minerals were being discovered in every corner of that large country, and he thought probably within the next few years coal would be found in so many places that Canada would be in the position of being practically independent of peat as a fuel.

Peat was never used in Canadian locomotives. At the present time in large districts of Canada oil-fuel was being used in preference to coal fuel for various reasons, not from a strictly economical standpoint so much as from the necessity of saving the forests from fires. Large areas of forests had been burnt, through sparks from locomotives, and enormous amounts of valuable timber had thus been lost to the world. He could only hope that, in the future, Canada would have still more of its timber wealth saved to the country by the use of liquid fuel, and also by enforcing on hunters and prospectors the exercise of more care in extinguishing their camp fires instead of leaving them to smoulder, when the first gust of wind would fan them into dangerous and destructive forest fires.

Mr. H. V. Pegg, in reply, said he thought the discussion had proved exceedingly useful; in fact, there were so many points upon which he would like to dwell that he was afraid he would not be able to complete his reply within the time allotted to him. Nature had been very lavish to Ireland in the matter of its bog supply. He had been over nearly every bog in Ulster, and in traversing those bogs one could only marvel at the change that would undoubtedly be wrought in the country if the bog-lands could be utilized. The bog-lands had been there from time immemorial, and the amount of time and money that had been, and was being, spent in trying to utilize peat for commercial purposes was enormous. Mr. Malcolm, in the course of his remarks (page 793), agreed with him in regard to the use of peat without treatment. Personally,

(Mr. H. V. Pegg.)
he went further, and would make bold to say that it was the only solution. A very large amount of money had been, and was being, spent on a process for preparing the peat, a matter that had been touched on by Mr. Dixon (page 796). He presumed that gentleman was referring to the Dumfries plant, which had been in existence a long time, but he had yet to hear that it was a success. It might be, and he would be only too glad to have information to that effect. The colossal amount of money that had been spent on preparing peat would be sufficient he thought to deter anybody scheming any further. He had a very fine briquette of peat which was as hard as wood, which burnt well, and which was exceedingly useful as a fuel, but the high cost of preparing it prevented its adoption. It was produced by a Dublin man. Whether the Dumfries process would turn out any better briquettes was quite an open point.

Mr. Malcolm mentioned the question of the use of prepared peat in locomotive boilers. Many years ago, when he was on the Great Eastern Railway, he tried firing the boilers with coal, and he found that quite a bad enough job for a novice ; and he was very glad he would not be on the foot-plate of any locomotive when peat was being tried. At the present time a peat fire was used under the Stirling boilers in Mr. Hamilton Robb's works, and it was used simply because it was a commercial proposition. It cost less to use plain air-dried peat taken out of the stack outside than it did to use coal. He asked the firemen what kind of a job it was, and they replied that they did not want a harder job, that it was quite enough for them, in fact they could just keep going. The peat was brought to Portadown for the production of gas in the gas-producer plant, and having the peat there during the recent coal strike, it was used under the boilers simply as an experiment. If it had not paid, its use would have been dropped, but it had been found that burning rough air-dried peat was better and cheaper than coal. But the main use for the peat was to turn it into water-gas in the gas-producing plant.

Mr. Malcolm had referred to the difficulty of preparing the peat on the bog-land, owing to the necessity of following up the peat. He had in mind a relic of one of the peat-preparing machines that

was standing on one of the bog-lands when he was there about three years ago. It was rapidly dropping into decay, simply because the cost of moving the machines up to the peat was more than the value that would be obtained by the superiority, if any, of machine working over hand working.

Mr. Patchell (page 794) had raised a question of the percentage of hydrogen in the analysis during the winter months. Unfortunately he had no ultimate analyses of the sodden peat, that is, peat with about 70 per cent. or more of water. The analysis (page 793) was obtained with normal air-dried peat, and 26 to 30 per cent. was about the figure when normal air-dried peat was used. With regard to the amount of hydrogen, he had mentioned in the Paper that in his own previous experiments the chief difficulty experienced with the plant installed in the Manchester district was the high thermal value of the gas, which was owing to the high percentage of hydrogen. It very nearly wrecked the engine because it was liable at any moment to exceedingly violent pre-ignition charges. The plant was simply kept running for the reason that they wanted to see what could be done with it, and as the engine was an old one it had to take its chance.

His experience was not quite the same as that of Mr. Andrews on the question of centrifugal extractors. He found that it was practically impossible by any means to get the water out of the peat, and that was the reason why it must be air-dried, in other words left alone. If an attempt was made to get the water out, troubles immediately began.

*Communications.*

Mr. FRANK FIELDEN (Chief Engineer, Gas Plant Department, Messrs. Crossley Brothers) wrote that, although the results obtained at Mr. Robb's factory were extremely good, a much better result could be obtained with a plant sufficiently large to justify ammonia recovery. The amount of nitrogen obtained in the recorded analysis, namely, 0·97 per cent., was extremely low for peat fuel. His firm had actually obtained analyses amounting to 2·7 per cent. from Irish peat, although it should be stated that there was a great variation even from the same portion of the bog. It was surprising and regrettable that developments in this direction had been hitherto confined to the Continent. Messrs. Crossley Brothers were now in a position to put forward a design of suction-gas plant which would not only successfully gasify peat, but which would, without any alteration to the generator, gasify waste fuels, etc.

Mr. A. BASIL WILSON and Capt. H. RIALL SANKEY (Member of Council) regretted that they had been unable to be present at the verbal discussion. They had since had an opportunity of considering the Paper, and hoped that the following remarks might be useful to the members.

It should be borne in mind that when any fuel containing moisture was burned under a boiler, a proportion of its combustible substance was used firstly to evaporate the moisture, and secondly to superheat it to the temperature of the flue gases; whereas if such fuel were used in a producer, a considerable quantity of the water contained in it could be used to cool the issuing gases, instead of carrying out this necessary process in a scrubber. They had no definite data as to how much water in peat could be dealt with by the producer without reducing its thermal efficiency. They considered it probable that peat containing 60 per cent. of moisture, representing 1·33 ton of water per ton of peat substance, would be about the practical limit.

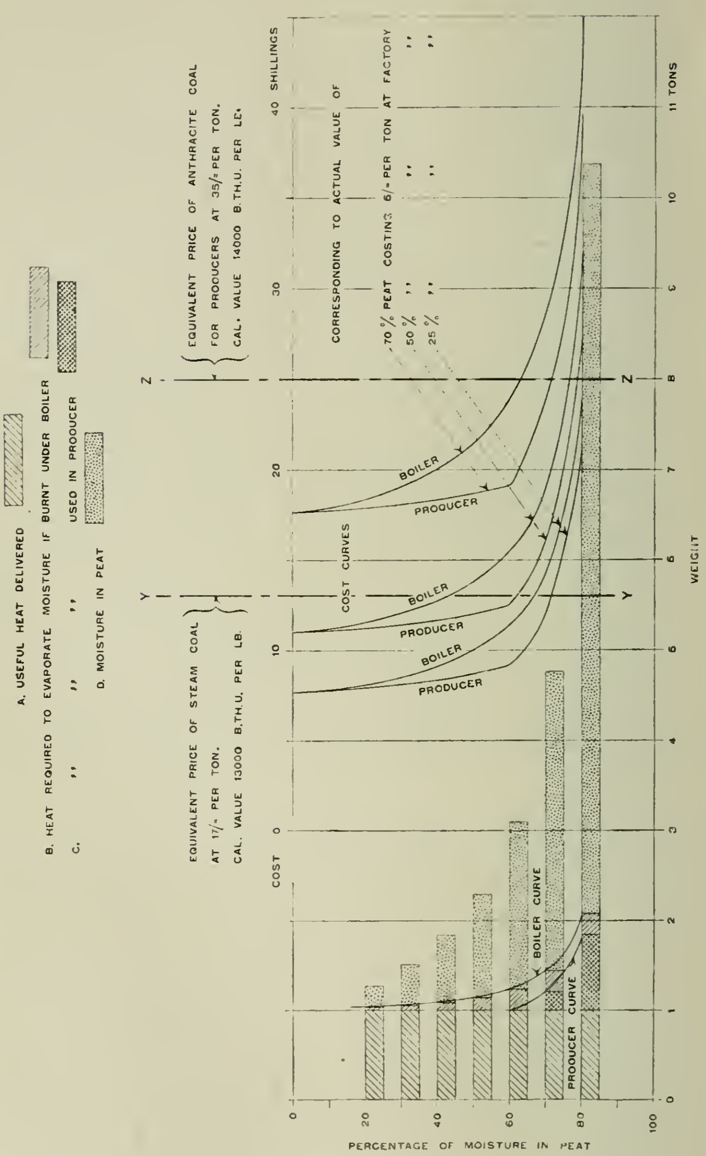
The diagrams shown in Fig. 1 (page 804) had been based on these considerations, in which the percentage of moisture in peat was plotted vertically and the weights horizontally. The equal rectangles (A) on the left represented 1 ton of peat substance (that is, peat free from moisture). The rectangles of varying lengths (B) represented the additional weight of peat substance required to evaporate and superheat the water contained in the various proportions. The top corners of these rectangles lay in the curves marked "boiler" and "producer" respectively. For a producer the areas were cross-hatched (C). The further row of rectangles (D) represented the weight of water in peat of various percentages of dryness when the peat was burned under boilers. In the case of peat used in producers, it was only when the moisture exceeded 60 per cent. that an extra weight of peat substance was required for evaporation, whereas if burned under boilers the demand for additional peat substance existed with all percentages of moisture.

It would be noticed how rapidly the weight of water increased at the higher percentages, and, incidentally, it might be pointed out that defining the quality of peat as a percentage of water-content was liable to cause misunderstanding; between 20 to 30 per cent. the increase in weight of water was comparatively small, namely 0·18 ton, whereas for a similar increase of 10 per cent. from 60 to 70 per cent. the added weight of water was considerable, namely 0·83 ton.

Referring to the Table (page 791) of comparative costs of Coal and Peat for running the factory, it was not stated what the water percentage in the 50 tons of peat was, but, on inquiry, the author informed them it was approximately 70 per cent. If so, and allowing one shilling per ton for carriage from the bog to the factory, the cost of peat substance could be shown to be 17·67 shillings per ton, and on this basis the cost-curves for peat containing various percentages, namely 25, 50 and 70 per cent., burnt under boilers and in producers, shown in Fig. 1, had been drawn. The vertical lines marked YY and ZZ showed the comparative cost of steam-coal and of anthracite to produce the same heating effect as one effective ton of peat substance, assuming

(Mr. A. Basil Wilson and Capt. H. Riall Sankey.)

FIG. 1.—*The Effects of Varying Amounts of Moisture in Peat.*



the calorific values of anthracite 14,000, steam-coal 13,000, and peat substance 10,000 B.Th.U. per lb. It would be seen that when the cost of 70 per cent. peat was six shillings per ton, the cost of burning under boilers was considerably greater than that of coal; but, when used in a producer, even up to 72 per cent. moisture, it was less than that of coal. If, however, six shillings was the cost per ton at the factory of 25 per cent. peat, a different set of cost-curves was obtained, as shown in the Fig. The "under-boiler" curve cuts the steam-coal line at about 65 per cent., and the "in-producer" line at 75 per cent. It might be remarked, however, that it was hardly possible for such high percentages as 75 per cent. to be effectively used either in boilers or producers.

The above indicated that some mistake had been made by the author in the statement that 70 per cent. peat was used in the comparison above referred to, and this was confirmed by the fact that the calorific value of $8\frac{1}{2}$ tons of anthracite, and 19 tons of steam-coal, worked out to $8 \cdot 2 \times 10^8$ B.Th.U.; whereas, taking 10,000 B.Th.U. per lb. as the calorific value of peat substance, the value of 70 per cent. peat was only $(1-0 \cdot 7)$ 10,000, or 3,000 B.Th.U. per lb. The value, therefore, of 50 tons was $3 \cdot 4 \times 10^8$, but a large proportion of this heat was necessary for evaporating and superheating the contained water, as had already been pointed out. In the Paper it was stated that approximately half the peat was used under boilers and half in producers. It could therefore be shown that under these conditions 12 tons of peat had to be burned for evaporating and superheating purposes, leaving only 38 tons effective. The calorific value of 38 tons of 70 per cent. peat being only $2 \cdot 55 \times 10^8$, there was obviously a large deficiency. If the question were asked: What percentage of water might there be in peat, so that 50 tons would produce $8 \cdot 2 \times 10^8$ B.Th.U.? a simple calculation would show that it must be 26. The cost-curves for 25 per cent. peat were therefore approximately applicable in such a case. It would seem, therefore, desirable that the author should ascertain what the quality of the peat was as used in comparison with coal as given in his Paper.

Mr. H. V. PEGG wrote that, in further reply to Mr. Andrews in regard to the preparation of peat, it was significant that the process mentioned by Mr. Andrews was not successful owing to the very reason mentioned by the author, namely the cost of treatment.

As regards the use of the tar for making tar-macadam for roads, the term "tar" was perhaps misleading, as it was by no means a pure tar or any approach to it. It would perhaps be nearer to call it an emulsion of oil and water, containing a considerable amount of solid matter consisting of very fine particles of charred peat. The amount of water in the so-called tar would probably be against its use for road-making, some of the samples analysed showing as much as 50 per cent. of water.

Mr. Dixon gave some very interesting information in regard to the production of by-products from peat; the author would be glad to know what it actually cost to recover the said 17 shillings' worth of sulphate of ammonia. The question of by-products recovery, generally, was a very large one. In the first place, ammonia recovery could only be considered on a very large scale. Secondly, a very large amount of steam was necessary for the process, so that by the time this had been generated very little gas was left over for power purposes. It was therefore doubtful if the recovery of by-products was of advantage, except in cases where the nitrogen content was high. The author had hoped that some figures would be forthcoming at the Meeting in regard to the peat plant in the South of England mentioned in the Paper (page 791), as that plant was an ammonia recovery plant and the results obtained could then be compared with those mentioned by Mr. Dixon. It would be interesting to know if this plant was still running and if it was commercially successful. It would be noted from the analysis of the sample of Portadown peat that the nitrogen content was low, namely 0·97 per cent. This was decidedly too low to make the question of ammonia recovery feasible, but it must be remembered that the peat in question was practically all surface peat. The nitrogen content of the peat increased with the depth of the bog; this was shown by analyses taken at varying depths.

Mr. Hill also suggested the use of the tar for spraying the roads, which question had already been replied to. As regards the nature of the tar from peat, it was presumably a vegetable compound, though the analysis showed crystals of paraffin wax to the extent of 1·26 per cent. of the original tar.

Mr. Lacey Johnson mentioned (page 799) that a great many experiments had been made in Canada in regard to the preparation of peat from some of the boggy districts; these experiments apparently have met with no better success than their predecessors. Canada with her huge resources would probably find all the fuel wanted without having to resort to peat. As regards its use in locomotives, peat was manifestly a most unsuitable fuel owing to its great bulk in proportion to its thermal value, and the first requisite for fuel for locomotives was portability.

The joint contribution on the part of Mr. A. Basil Wilson and Captain Sankey was of the greatest interest, and the author had studied the diagrams and curves with great pleasure. From the producer cost-curves, it was evident that the critical point was reached with 60 per cent. of moisture; after that point the curve extended rapidly. The chief difficulty that arose was the deciding of the precise percentage of the water contained in the peat, as not only did this percentage vary in individual peat-blocks in any given lot, but it also varied in individual blocks from day to day owing to weather conditions, and this consideration probably accounted for the mistake mentioned. The 70 per cent. peat referred to was approximately the percentage of water contained in the peat when it was fired into the producer, but as this peat had been lying on the ground exposed to drenching rain-storms for a considerable period before being fired into the producer, the percentage of water was naturally higher than when it was delivered on site. The 50 tons of peat mentioned was the weight of peat as loaded on to the barges at the bog-lands, and at that time the percentage of water was probably very much nearer the figure given by Mr. Wilson and Captain Sankey, namely 26 per cent., which was somewhat about the over-all average that might be expected in normal air-dried peat.

## EXCURSIONS.*

On TUESDAY AFTERNOON, 30th July, the Members proceeded by special tramcars to the Works of Messrs. Harland and Wolff, Queen's Road, where they were entertained at luncheon by the Directors of the Company, under the chairmanship of Mr. John W. Kempster. Subsequently they were conducted over the Works and Shipyards by members of the staff.

At Five o'clock the Belfast Corporation Fire Brigade arranged a special turnout at the Headquarters, Chichester Street, under the direction of Mr. George Smith, Superintendent.

In the evening the Institution Dinner was held in The Ulster Hall, by permission of the Corporation of Belfast, the Hall having been tastefully decorated by the Reception Committee. The President occupied the Chair, and 225 Members, Ladies, and Visitors were present. Among the latter were the Members of the Reception Committee, which included The Lord Mayor, Councillor R. J. M'Mordie, M.A., M.P., *Chairman*, and the Lady Mayoress; Mr. J. Milne Barbour, D.L., and Mr. Robert Thompson, D.L., M.P., *Vice-Chairmen*; Mr. H. Incledon Johns, *Treasurer*; Mr. S. C. Davidson and Mr. Bowman Malcolm, *Joint Honorary Secretaries*; Mr. James Mackie, *Chairman of Executive Committee*; and Mr. J. G. Harris, *Assistant Honorary Secretary*.

The President was supported by the following Members of the Council of the Institution: *Past-Presidents*, Mr. John A. F. Aspinall, Mr. William H. Maw, LL.D., and Mr. J. Hartley Wicksteed; *Vice-President*, Mr. Michael Longridge; *Members of Council*, Dr. H. S. Hele-Shaw, F.R.S., Mr. Robert Matthews, and Mr. William H. Patchell.

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* The notices here given of the various Works, etc., visited in connection with the Meeting, were supplied for the information of the Members by the respective authorities or proprietors.

After the Loyal Toasts had been honoured, Mr. JOHN A. F. ASPINALL (Past-President) proposed that of "The City and Industry of Belfast," and said that the Lord Mayor had reason to be proud of the city, and the citizens had good cause of pride in their Lord Mayor, who had filled the office with great distinction for three successive years, while he was also the representative of one of the important divisions of the city in Parliament. There was no doubt that the trade of Ireland had been improving enormously in recent years. The figures of the Department of Agriculture and Technical Instruction showed that for the five years ending 1910 the exports from Ireland amount to £65,800,000. For the first time they had exceeded the imports by £800,000. The exports of manufactured goods, in which Belfast took such an important part, had grown by £6,500,000. In Dublin, Cork, Waterford and Limerick they saw evidence also of advancing trade, and everywhere throughout the country it seemed to be the same. In Belfast they had some wonderful industries. The exports of their linen goods alone came to £13,000,000. That represented about one-fifth of the total exports of the country. That day they had the opportunity of seeing one of their great shipbuilding works—instituted years ago by Sir Edward Harland and Mr. Wolff. The export of shipbuilding from this country was over £3,500,000. Belfast was endeavouring to encourage shipbuilding, and had altered her river—deepened it, widened it—added docks and quite recently created a magnificent graving dock, which would accommodate the largest ships in the world. Every kind of encouragement was given to trade. He felt they could all congratulate the Lord Mayor and citizens of Belfast on what had been done for Ireland. Emigration was decreasing, the flow of trade was increasing by leaps and bounds, and at any rate the citizens of this great city had done what they could to find work for the sons of the soil.

THE LORD MAYOR, in acknowledging the Toast, said that long ago there was a considerable amount of manufacturing of woollen goods carried on in this country, and a great grievance had existed

because things were so regulated on the other side of the Channel that the trade was lost; but that had been forgiven. The men who had been engaged in that industry were driven to the United States of America, and they were the very men who secured the independence of the United States. That, however, was all past, and two years hence they would all be celebrating the hundredth anniversary of the declaration of peace between the United States and the British Empire. Referring to imports and exports, Belfast was far ahead of any district in England or Scotland. They were also far ahead in regard to unemployment, and so far as Poor Law relief was concerned they had 50 per cent. less than the average in England and Scotland. In his last report the American Consul stated that Belfast and the surrounding district contributed four-fifths of all the linen manufacture of the world. He was sure they were all pleased to know that the imports and exports of the city were growing, and that the average was much ahead of any other place on the other side of the Channel. The city was peculiarly pleased with the visit of the Institution of Mechanical Engineers. Other associations visited them from time to time, and the citizens were always glad to welcome them, their desire being to make every one feel at home and enable them to have a pleasant and profitable visit to this very energetic community.

Mr. R. Garrett Campbell, who also responded, said he thought it would not be seemly if some member of the industrial community of Belfast did not thank the Mechanical Engineers for the way in which they had received that toast. Belfast manufacturers, in common with all other manufacturers, recognized that they owed to the Mechanical Engineers a deep debt of gratitude. Without their assistance there would be no such thing as industries in the modern sense of the term, and they would be living under the conditions of the past to which Mr. Horner had alluded in his Paper on spinning-wheels and distaffs. Mechanical Engineers had made it possible to organize industry on a large scale.

Mr. J. Hartley Wicksteed (Past-President) in proposing the Toast of "The Reception Committee," remarked that sixteen years

(Mr. J. Hartley Wicksteed.)
ago the Institution came to Belfast, and they then had a most hospitable Reception Committee. On the present occasion they had a very large and influential Committee, to whom the thanks of the Members were due, and he could not allow the occasion to pass without mentioning their obligations especially to the Honorary Secretaries, Mr. S. C. Davidson and Mr. Bowman Malcolm. He coupled the Toast with the names of Mr. Robert Thompson and Mr. J. Milne Barbour, Vice-Chairmen of the Reception Committee.

Mr. ROBERT THOMPSON, D.L., M.P., in replying, said he felt assured that the Reception Committee had endeavoured to discharge efficiently the duties entailed upon them by this highly appreciated visit to the loyal, industrial city of Belfast. The Reception Committee were sensible that a high honour had been conferred on this city by that timely visit of the honoured and distinguished members of the Institution of Mechanical Engineers, who could boast of many noble works well and faithfully done, in which the sons of Britain had achieved great and illustrious distinctions. If they went back about three centuries before the Christian era they found Archimedes had then distinguished himself as the greatest mathematician and engineer of which they had any historical record, but they must come to more modern times to find in the seventeenth century that Sir Isaac Newton discovered the laws of gravitation and of motion, force and energy, and provided formulæ by which the earth and the planets could be measured and weighed as in a balance. A century later they came to James Watt, the inventor of the steam-engine, and almost concurrently Richard Arkwright invented the spinning-jenny. A little later came George Stephenson's inventions. During their own time they had carefully watched the progress made in the steam-engine, and in more recent times they found the invention of Morse's telegraph introduced, by which, with improvements, they had ultimately encircled the globe. In 1838 the first steamship crossed the Atlantic, and the vessels had now developed into monster ships such as the "Olympic" and the unfortunate "Titanic," which deserved a better fate. In 1877 they had Bell's

telephone introduced, which was now indispensable, not only in business but also in private houses. In 1878 came Edison's important discoveries in electric light. In 1883 they had the first electric tramway in Great Britain opened at Portrush, leading to the Giant's Causeway; in their own times they had had motor vehicles introduced, which to-day monopolised the streets of all their large cities, and still more recently they had had the introduction of aeroplanes, monoplanes, and hydroplanes, all of which bade fair to revolutionize locomotion, not only in times of peace but in times of war. Such were some of the distinctions they owed to skilled and inventive engineers, whom they regarded as the pioneers of works of supreme importance. They as members of the Reception Committee bade them a right hearty welcome to the city of Belfast.

Mr. J. Milne Barbour, D.L., who also responded, pointed out that Belfast had grown from village proportions to one of the great cities of the United Kingdom in the course of about a century, and when entertaining members of a profession which had done so much to contribute towards that development they could not but acknowledge the debt they owed to mechanical engineering in the three great branches which had been referred to—namely, introduction of mechanical power, means of transport, and means of rapid communication of intelligence. In all those three great factors, which had revolutionized commerce, mechanical engineering certainly could claim the greatest credit. They in this country would cherish the hope that those who had done so much to curb the rude violence of disruptive forces, and convert them into useful and well ordered energy to the aid of industry and progress, would be able to achieve yet other victories in that direction, and possibly not limit the scope of their activities alone to mechanical engineering.

The Toast of "The Institution of Mechanical Engineers" was proposed by Mr. James Mackie, Chairman of the Executive Reception Committee, who congratulated the President on the

(Mr. James Mackie.)

important position which the Institution of Mechanical Engineers occupied in the world of Engineering. When he looked at the illustrious names of their Presidents he scarcely wondered that it had had such progress—names which were associated with everything that had done so much for England and for the world—and he was proud as a Britisher to think that in the development of engineering England took the most prominent place. When he met the engineers of other countries he found they were men with a knowledge of banking, of commercial work, of balance sheets, knowledge of other countries, and possessing the power to speak in the languages of other countries. It seemed to him that if British engineers were to progress in the future as they had done in the past, it would be necessary for the Institution to consider that point of education to which he had just referred. It appeared to him sometimes that as engineers they had not lived up to all their opportunities. Finance was controlling engineering instead of engineering controlling finance. Speaking to the younger men, he said that what they had to do was to keep their aims high and to think big, and if they did that they would come out all right in the end. He had great pleasure in submitting that Toast.

The President, in responding, said that the Proposer of the Toast had placed before them as mechanical engineers a very high ideal. He had spoken feelingly of their difficulties and of their aspirations, and he was sure he expressed the Members' opinion when he said that it was the strong desire of the Institution of Mechanical Engineers to help forward that ideal which he had so nobly expressed. Mr. Thompson had referred to the distinguished line of Past-Presidents of that Institution, and he had mentioned the name of George Stephenson. They must never forget that it was their proud boast that George Stephenson was their first President. Mr. Thompson had also referred to other distinguished men, including Lord Kelvin, who was born in Belfast, and, although he left this country in his youth, the work that he did penetrated all over the world, and Belfast had benefited as much as any city

from his labours. They were aware it was proposed to place in Westminster Abbey a window in memory of Lord Kelvin, and in that matter the Institution of Mechanical Engineers was taking part. During the last fortnight the Royal Society of England had been celebrating its two hundred and fiftieth anniversary, and it was a coincidence that the birth of the prosperity of Belfast dated from about the same time. Ideas were then very different from what they were now, and though the Royal Society was associated with many curious experiments at that time there was no conception that it would attain to the position it occupied to-day. Reference had been made to the knowledge and experience required by mechanical engineers, and he heartily sympathized with the idea that engineers should not confine their attention entirely to engineering, but they ought to be broad-minded and cultured men. It was largely with that view that they were instituting a test of such general knowledge as was implied in their examinations. Reference had been made to the work of the Reception Committee, but he could not sit down without reminding them that the work of those outings largely depended upon the staff of the Institution, with Mr. Worthington as their secretary. He hoped the rest of their visit might be as auspicious as the proceedings that day had been.

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On Wednesday Afternoon, 31st July, after luncheon in The Ulster Hall, by permission of the Corporation of Belfast, a large party of Members and Ladies went by special steamer, kindly provided by the Harbour Commissioners, to visit the Harbour Works and the new Graving Dock. At the latter place refreshments were provided in a marquee by kind invitation of Mr. Robert Thompson, M.P., Chairman of the Harbour Commissioners, who, with several of the Commissioners and Officials, accompanied the party on the trip. A short excursion was afterwards made on Belfast Lough.

Individual visits to the other Works (page 817) were also made during the afternoon.

In the evening a Reception was held in the City Hall by The Lord Mayor and Lady Mayoress.

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On Thursday Morning, 1st August, visits were made to the Engine Works and Shipyards of Messrs. Workman, Clark and Co., and to one or other of the Textile Works named on page 817.

The Weaving Works and Peat Fuel Plant of Messrs. Hamilton Robb at Portadown were also visited by a number of Members.

In the afternoon Mr. and Mrs. S. C. Davidson gave a Garden Party in the grounds of their residence at Seacourt, Bangor, the Members and Ladies, to the number of about 200, making the journey by special train. During the afternoon the Band of the King's Own Scottish Borderers played in the grounds.

In the evening the Members and Ladies attended a special Cinematograph Exhibition of subjects of engineering interest in the Picture House, Royal Avenue, by invitation of the Directors.

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On Friday, 2nd August, two whole-day Excursions were made.

One was to Portrush by special train, kindly provided by the Midland Railway Company. After luncheon in the Railway Station Dining Room, the party, consisting of Members and Ladies, proceeded in special electric cars to Dunluce Castle, over which they were conducted by Mr. William A. Traill, M.A. Ing., who described its history. The journey was then resumed to the Giant's Causeway, where Mr. Traill again guided the Party and explained the geology of the district. After tea in the Chalet at the Causeway, the return journey was made in special cars to Portrush, where dinner was served in the Railway Station Dining Room.

The second Excursion was made to Newcastle, Co. Down, by special train, kindly provided by the Belfast and County Down Railway Company. On arriving at Newcastle the Members and Ladies proceeded by carriages to Bryansford, where they inspected

the gardens and demesne of Tullymore Park, by kind permission of the Earl of Roden. After luncheon in the Slieve Donard Hotel, Newcastle, the party visited Donard Park and the Golf Links.

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The following additional Works were open to the Members during the Meeting:—

David Allen and Sons, 16 Corporation Street. (Printing and Publishing.)
Belfast Central Public Library, Royal Avenue. (Spinning Wheels, illustrating Mr. John Horner's Paper, page 685.)
Belfast Corporation Electricity Works, East Bridge Street.
Belfast Corporation Gas Works, Ormeau Road.
Belfast Corporation Tramways, Car Works and Engineering Department, Napier Street.
Belfast Municipal Technical Institute, Mechanical Engineering Laboratory, College Square East.
Cantrell and Cochrane, Victoria Square. (Aerated and Mineral Waters.)
Dunville and Co., Royal Irish Distilleries, Distillery Street.
Inglis and Co., 35 Eliza Street. (Bread and Cake Manufactory.)
Belfast Ropework Co., Connswater.
Broadway Damask Co., Broadway.
Brookfield Linen Co., Crumlin Road (Flax and Tow Spinning); Cambrai Street (Weaving, Bleaching, Dyeing and Finishing); and Agnes Street (Weaving).
William Ewart and Son, Crumlin Road Mills. (Flax and Hemp Spinning and Weaving.)
Gunning and Campbells, North Howard Street, Falls Road. (Flax and Tow Spinning.)
York Street Flax Spinning Co., York Street.

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During the Meeting the Members were invited to be Honorary Members of the following Golf Clubs: Royal Belfast Golf Club, Carnlea, Crawfordsburn, Co. Down; Royal County Down Golf Club, Newcastle; and Royal Portrush Golf Club, Portrush.

# BELFAST CORPORATION ELECTRICITY GENERATING STATION.

The Central Electricity Generating Station is situated at the corner of East Bridge Street and Laganbank Road, and provides not only the lighting and power of the City, but also the whole of the energy for the City Tramways. The No. 1 Engine Room contains 3,800 kw. of plant, which is used exclusively for the supply of electricity for lighting and power. In 1905 the building was extended in order to meet the increased demand due to the electrification of the tramways.

The No. 2 Engine Room contains three 1,000 kw. steam dynamos. The engines were built by the local firm of Messrs. Combe Barbour and Combe, Ltd., and the dynamos by The British Westinghouse Co., and are used for the tramway supply only. There are also two 1,000 kw. turbo-dynamos, and one 1,500 kw. turbo-dynamo for the lighting and power supply. The turbines were built by Messrs. Willans and Robinson, of Rugby; the 1,500 kw. machine is one of their latest combined impulse and reaction type. Two of the dynamos are by Messrs. Brown, Boveri and Co., and the third by Messrs. Siemens Bros. The total capacity of the combined station is 10,300 kw.

The system of supply is continuous current at 220 and 440 volts pressure for lighting and power purposes, and 550 volts pressure for tramway purposes. A portion of the supply is converted, by means of motor generators, to 3-phase alternating current at 6,000 volts pressure for transmission to the Fortwilliam Sub-Station, where it is again converted to continuous current for the tramways and lighting supply in that neighbourhood.

The boiler-houses contain six Lancashire and fifteen Babcock and Wilcox water-tube boilers; the latter are fitted with mechanical chain-grate stokers and fuel economisers, together with auxiliary

machinery. A conveying plant is provided, so that the coal can be brought alongside the quay wall at Albert Bridge and delivered mechanically either into the bunkers over the boilers or into a group of storage bunkers (1,600 tons capacity) situated adjacent to the boiler-house, the ashes being removed on the returning half of the conveyor. The pump-house, situated on the riverside, where electrically-driven centrifugal-pumps are installed, provides the necessary circulating water for condensing purposes. Mr. T. W. Bloxam is the City Electrical Engineer.

## BELFAST CORPORATION FIRE BRIGADE.

The Belfast Fire Brigade is constituted under the Belfast Local Act, 1845, and it was originally worked by, and in conjunction with, the Town Police Force with about twenty men and one station, and afterwards by a small permanent and auxiliary staff until the year 1892. In that year the Brigade was reorganized and strengthened, and has been further augmented from time to time, and now consists of five stations and eighty-one men, thirteen horses, four motors and the various fire-extinguishing equipment, thirty-nine street fire-alarm call points, and a number of automatic fire-alarms.

The headquarters station in Chichester Street was erected and opened in 1894; the site is practically square, being 70 yards by 65 yards, having a frontage in Chichester Street, Oxford Street and Town Hall Street, and adjoining the police cells and police offices on the other side. The station is of three floors, and fronts on to Chichester Street, and consists of engine-house, offices, stables, gymnasium, observation tower, etc., men's dwelling-houses of two floors abutting on to Oxford Street and Town Hall Street, with engine- and boiler-house, stables, hay-lofts, etc., adjoining the police cells, and a drill-yard in the centre of the site.

The branch stations were erected and opened as follows, viz.: Spiers Place Station in the year 1887, Whitla Street in 1898, Albert Bridge Road in 1904, and Ardoyne in 1904.

Since the year of the reorganizing of the Brigade an ambulance service for accidents, etc., has been worked by the Brigade, and the staff have been successful in winning cups and prizes in competitions under the St. John Ambulance Brigade regulations. The Brigade is under the control and managed by a Committee of the Corporation. The Chief Officer is Mr. George Smith.

## BELFAST CORPORATION GAS WORKS.

The Belfast Gas Works are situated close to the River Lagan, and are hemmed in between the Great Northern Railway, the Ormeau Road and private property. A Parliamentary Bill is now being promoted to enable the Corporation, among other things, to acquire the latter property for purposes of extension.

These works were first established in 1823. The ground occupied was only about 1½ acre, and the plant was necessarily on a very small scale. The concern does not seem to have been a very prosperous one, as for thirty years or so many difficulties had to be met, but from 1852 onwards business improved greatly, and when the Corporation purchased the undertaking in 1874 it was a great financial success.

The output of gas has continued steadily to increase, as will be seen from the following Table:—

| | | |
|---|---|---|
| 1875 | 418 | millions of cubic feet. |
| 1880 | 548 | ,, ,, ,, ,, |
| 1885 | 718 | ,, ,, ,, ,, |
| 1890 | 978 | ,, ,, ,, ,, |
| 1895 | 1,169 | ,, ,, ,, ,, |
| 1900 | 1,305 | ,, ,, ,, ,, |
| 1905 | 1,816 | ,, ,, ,, ,, |
| 1910 | 2,236 | ,, ,, ,, ,, |

The output for the year ending 31st March last reached 2,338 millions, and, as far as can be judged, the possibilities are enormous, on account of the numerous uses to which gas is now put for heating and power purposes.

All the coal-gas is produced at present in horizontal retorts, but, under a new scheme of extensions proposed by the present engineer, these will be displaced by verticals in the near future. The first section of these is already in course of construction in the works. The vertical retort adopted is the Glover-West system of continuous carbonization, and among the advantages claimed are: small amount of space, simplicity of mechanical parts, cleanliness in working, and freedom from nuisance. The present carbonizing plant will be gradually done away with, and the vertical retorts arranged in $2\frac{1}{2}$-million sections until the scheme is completed.

The method of carbonization is as follows: The coal, having been supplied to the coal-feeding hopper above each retort, slowly and continuously falls by gravitation to the retorts, and through the same to a coke-receiving chamber; the coal, passing on through the retort, becomes completely carbonized, and the residual coke is extracted by means of a worm extractor at the base of each retort. The speed of the worm regulates the speed at which the coal is continuously passing through the retort. The coke accumulated in the coke-receiving chambers below the retorts is periodically discharged to the coke-conveyors.

In connection with this system of carbonization the heating of the retorts is an important feature, and is specially applicable to continuous carbonization in vertical retorts. The combustion-chambers are arranged in tiers, each chamber being separated from the other. The products of combustion, after passing around the retorts, enter a common duct which is interior to the combustion-chambers, and the waste gases circulate through chambers situated at the top end of the retorts before entering the chimney. Further, the secondary air supplied for completing the combustion of the producer-gas in the several separate combustion chambers circulates around the base of the retorts, extracting the heat transmitted from the coke through the walls

of the bottom section of the retorts which are at the base constructed of cast-iron. The recovery of the heat from the coke ensures very considerable economy in fuel consumption; it also enables the coke to be discharged in such a cool state that no quenching is required. These are decided advantages with this system, and they enable the attendants to carry out their duties, under which conditions they are not inconvenienced by steam or fumes arising from the coke as in the ordinary system where coke has to be quenched. The working of the plant is noiseless and free from dust, flame, and smoke, so common with other systems for the carbonization of coal.

The construction of the coke-extractor is another important point in connection with the Glover-West system, as the worm is constructed in two parts to enable one part to revolve away from the other, and discloses an aperture sufficiently large to readily permit the use of scurfing tools and the retort to be scurfed. The extractor-gear will be arranged with a considerable range of speed regulation which will enable the speed of the coal passing through the retort to be regulated according to the class of coal being carbonized. A safety device is fitted to each set of coke-extractor gear to prevent breakage in the event of an overload.

It is proposed eventually to build on adjoining ground a large gas-holder capable of holding $7\frac{1}{2}$ to 10 million cubic feet, but in the meantime No. 5 Holder has been demolished and a new four-lift telescopic spiral-guided gas-holder is in course of erection in the old tank. The contract for this work has been given to Messrs. Robert Dempster and Sons, of Elland, and includes, in addition to erection of new holder, the demolition of No. 1 and No. 5 Holders, and the erection of existing crown framing of No. 5 Holder in the tank of same. No. 5 Holder, which has been removed, was a two-lift telescopic one, each lift being 32 feet 3 inches deep. The inner lift was 176 feet in diameter and the outer 180 feet. The dimensions of the gas-holder to be erected are as follows:—

Inner Lift 171 feet 9 inches diameter × 33 feet 3 inches deep.
Second Lift 174 feet 9 inches diameter × 33 feet 3 inches deep.

Third Lift 177 feet 9 inches diameter × 33 feet 3 inches deep.
Outer Lift 180 feet 9 inches diameter × 33 feet 3 inches deep.

This work, as well as the vertical retort bench, will be well advanced by the time of Meeting in July–August. Mr. James D. Smith is the Engineer and Manager.

## BELFAST CORPORATION TRAMWAYS: CENTRAL DEPOT AND CAR WORKS.

This depot, which was reconstructed in 1906, is situated in Napier Street, Belfast. The Central Car Shed is entered from Gaffikin Street by twenty-one gates, hung to cast-iron stanchions. Its area is about 30,000 square feet, and it has room for 69 cars. On the floor above the shed wood-working machinery is installed; formerly it was occupied by the horses in a series of stables.

The Car-Body and Painting Shop was originally a three-storey building, the ground floor being used for cars, the first floor for hay, and the second floor for grain. The present Car-Building Shop is 250 feet long by 42 feet wide; it was formed by the removal of the two floors, the roof being supported by compound girders, this giving a clear space of over 10,000 square feet, in which three lines of rail are laid.

The Machine and Truck Shop occupies considerable space on the Napier Street side of the building. It has excellent lighting, both from the roof and side windows, whilst artificial lighting is supplied by enclosed arc lamps and incandescent lamps. The Blacksmiths' Shop adjoins, thus facilitating forgings being easily delivered to the machines. The cars enter by a single line, which branches out into two lines, running to overhauling pits. In the re-wheeling of cars a new method has been introduced by the Chief Engineer, Mr. A. A. Blackburn. Sections of the rail over the pits can be removed, and the wheels lowered by means of a hydraulic jack; they are then removed to the end of the pit ready for lifting.

Overhead is a large travelling crane, which runs the whole length of the shop and can lift up to $7\frac{1}{2}$ tons.

The top corner of the Machine Shop is devoted to a large and well-arranged Tool room. The shop generally is fitted with modern and up-to-date machinery. Special methods have been devised in many cases. Axle and armature bearings and brasses are turned out in a short space of time by special jigs.

At the bottom of the shop is a tyre-shrinking gas-furnace, designed by the Chief Engineer and built by the staff; it consists of two iron castings, bolted together, with the centres cored out to receive the ends of the wheel-axles, and through the centre is a trough, where the waste water is run off. Two $1\frac{1}{2}$-inch gas-rings, with 35 burners each, are fed with an air-blast, the supply being taken from the smithy fan. Over the press a travelling-crane runs parallel, and fitted to the wall is a semi-rotary pump which pumps up water for cooling the tyres. In this press, wheel-tyres can be heated up in 15 minutes ready for shrinking on.

In the Armature Room repairs are made to all the electrical appliances in connection with the car-equipment and establishment lighting. At one end of the shop is an electric oven, which is built entirely of concrete, and is capable of receiving four armatures at the same time, as well as field-coils and controller parts. At the other end of the shop is a water rheostat, which is built of hard-wood, and measures 3 feet square by 5 feet high. This tank is filled with water in which are immersed the electrodes, consisting of perforated iron cones. The rheostat gives a current variation of from 20 to 500 amperes at 500 volts.

The Smiths' Shop, 60 feet by 18 feet, occupies the space between the machine shop and the car-building shop, and is equipped to turn out all the necessary forgings used on the truck, car-bodies, and establishment work. It contains four forge fires, 4 feet by 4 feet, which are worked with a motor-driven fan-blast. At one end of the smithy a Babbitting plant is provided for refilling with white metal all bearings used on car equipments.

Adjoining the Machine Shop is the Car-Body Shop and Paint Shop. Twenty cars can be built or reconstructed at one time,

Fifty of the old horse-cars have been converted and put into service, being re-equipped according to the new standard, and, as far as possible, the standardization of car equipments has been observed.

The Paint Shop is situated at the far end of the Body Shop. On either side of the shop the car-bodies are placed in their order of advancement, and the various stages of priming, filling, rubbing down, colouring, decorating, and varnishing can be followed through each day.

The Wood-Working Shop is well provided with tools, including a 42-inch band-sawing machine by J. Sagar and Co., Halifax, a power mortising-machine with graduated stroke by Robinson and Son, Rochdale, a circular moulding-machine and self-acting saw-bench with rope-feed, also a combined hand and power-feed planing machine with superposed tables, all by Robinson and Son.

A School for Motormen has been established, and is now situated at the Sandy Row depot. There are seven depots in which to store the 300 cars, situated at various points of the system.

The Chief Engineer is Mr. Albert A. Blackburn, M.I.Mech.E., and the General Manager is Mr. Andrew Nance.

## MUNICIPAL TECHNICAL INSTITUTE: MECHANICAL ENGINEERING LABORATORY, BELFAST.

The Mechanical Engineering Laboratory, which was opened in November 1911, is situated on the ground floor of the Institute, and is 114 feet long by 42 feet wide. It is divided into two long bays and is covered with a glass roof. Two hand travelling-cranes run its whole length. The floor is double, there being a clear space of 3 feet between the levels, which permits the placing out of sight of all apparatus not directly needed. The

shafting is also placed below ground. Trap-doors afford easy access to this space. The laboratory is lighted by sixteen 200-c.p. drawn-tungsten lamps, and the underfloor by incandescent lamps.

*Boiler-House.*—The Lancashire boiler, 28 feet by 8 feet, was manufactured by Messrs. Joseph Adamson and Co., Hyde, and in connection with it is installed a Sugden superheater. The marine boiler, 11 feet by 11 feet, was made by Messrs. Workman, Clark and Co., Belfast. Inside the steam-space is a feed-water heater by Messrs. Hamilton and MacMaster, Belfast. There are two feed-pumps—one by Messrs. G. and J. Weir, Glasgow, and the other by Messrs. Pearn and Co., Manchester. Space is left in the boiler-house for a tubular boiler.

*Steam Section.*—The main steam-engine is of the horizontal cross-compound type, and was made by Messrs. Combe, Barbour and Co., Belfast. It is of 60 i.h.p., at a steam-pressure of 120 lb. per square inch, 150° F. superheat, and 100 revolutions per minute. It was designed for full boiler-pressure on the low-pressure cylinder. The high-pressure rods are made the same size as the low-pressure for the sake of symmetry. It can be used under any of the following conditions: (1) Compound condensing; (2) Compound non-condensing; (3) Single-cylinder condensing (using low-pressure cylinder); and (4) Single-cylinder non-condensing (using low-pressure cylinder). The engine is so arranged that the cranks can be set at any desired angle. The valves are all of the drop-piston type, the steam-valves being operated by a new form of trip-gear.

A 16-kw. Parsons turbo-alternator and exciter are installed. The alternator delivers three-phase current, which is taken to the main switchboard.

A high-speed engine of 25 b.h.p., at 750 revolutions per minute, by Messrs. W. H. Allen, Son and Co., Bedford, is installed. It is of the enclosed type, having forced lubrication, and drives directly a direct-current generator of 15 kw. at 220 volts.

A De Laval steam-turbine, by Messrs. Greenwood and Batley, Leeds, of 15 h.p. and 24,000 revolutions per minute, drives a centrifugal pump through a 6 to 1 gearing. The pump lifts 160

gallons of water per minute against a head of 160 feet. The various pressure-gauges enable an experimenter to carry out consumption tests; the nozzles can be easily withdrawn, and can be tested on a special apparatus of the Stodola type erected on the wall of the laboratory.

There are two condensers: (1) a surface condenser by Messrs. Isaac Storey and Sons, Manchester, and (2) an ejector condenser by Messrs. Ledward and Beckett, London. The former is of the contra-flow type.

*Main Switchboard.*—A Westinghouse double-panel marble switchboard is situated within 6 feet of the high-speed engine and turbo-sets. This controls the current from the two dynamos which form the loads for the high-speed engine and turbine. In order that students should learn various methods of testing, an Allen direct-current generator, running at 750 revolutions per minute, is attached to the high-speed engine, and a Parsons alternator to the turbine.

*Air-Compression Plant.*—The primary object of this plant is to study the efficiency of an ordinary two-stage air-compressor. This was made by Messrs. Alley and MacLellan, Glasgow, and is of their "Sentinel" type. The air-pressure generated is from 60 lb. to 80 lb. per square inch.

*Refrigerating Plant.*—This consists of a refrigerating machine and a liquid-air machine. The former was made by Messrs. L. Sterne and Co., Glasgow, and is employed to produce small quantities of ice, and to illustrate the method of cooling by means of a flow of brine.

The liquid-air plant was manufactured by the British Oxygen Co., London, and consists of a Whitehead air-compressor belt-driven from the underground shaft, an air-liquefier, and both low- and high-pressure purifiers. The gas-compressor is capable of compressing about 550 cubic feet of gas per hour to a pressure of 150 atmospheres or more when running at 400 revolutions per minute. Under these conditions it requires from 6 to 7 h.p. to drive it, and produces through the liquefier $1\frac{1}{2}$ litres of air per hour.

*Hydraulic Section.*—Various heads of water up to 280 feet are

obtained from a motor-driven high-lift turbo-pump, made by Messrs. Mather and Platt, Salford. It is used for pump-testing purposes, and supplying pressure-water to any one of three experimental turbines. It is of the three-stage type, and is capable of delivering 400 gallons of water per minute at a head of 200 feet. The pump is driven direct by a direct-current electric motor of 40 h.p., 440 volts, and 77 amperes.

The cast-iron channel, which receives the discharge water of the turbines, is of internal cross-section 2 feet 6 inches by 2 feet, and has an over-all length of about 30 feet. The tumbling bay may be fixed in three different positions corresponding to those required for discharging into either end of the large measuring tank or into both ends at the same time. A Venturi meter is used for measuring the quantity of water flowing from the high-lift turbo-pump. Pitot tubes are inserted in the pipe between the pump and the Venturi tube.

*Turbines.*—A Pelton wheel, made by Mr. Percy Pitman, London, develops 5 h.p. with a fall of 75 feet, when making 400 revolutions. The Girard turbine was made by Messrs. Gilbert Gilkes and Co., Kendal, and has a wheel of 21 inches diameter; it develops 6 h.p. on a 75-foot head when making 550 revolutions per minute. The Thomson turbine, also by Messrs. Gilkes and Co., gives about 4 h.p., with a fall of 75 feet.

*Testing of Materials Section.*—All tension, compression, and cross-breaking testing is carried out on a Riehlé automatic and autographic machine, which is capable of giving a maximum load of 150,000 lb. For experiments on heat treatment of various steels, a gas-furnace by Mr. S. N. Brayshaw, Manchester, has been installed, and for carrying out tension tests on cement an Adie machine is placed in this section. Tests on the fatigue of metals are made on a machine which was designed by Professor Smith and constructed by Messrs. Combe, Barbour and Co., Belfast.

*Internal-Combustion Section.*—The plant includes a gas-engine, a suction-gas plant, an oil-engine, and a petrol-engine. The gas-engine is by the National Gas-Engine Co., and is of 20 i.h.p. when working on ordinary town gas. The suction-gas plant was made

by the same company. The oil-engine is of the Blackstone type, and is of 12 i.h.p., and the petrol-engine is of about 4 h.p.

*Machine Shop.*—This is equipped with modern high-grade tools and machines, and is situated on the ground floor. Manual training metal-work classes are held in this shop, in connection with the Trade Preparatory Section of the Institute, and special classes are held in screw-cutting and milling for advanced students.

*Courses in the Department.*—These comprise (*a*) Day Technical Course; (*b*) Day Apprentices' Course; and (*c*) Evening Course.

In the Day Technical Course the instruction given is of University standard, and the programme extends over three years. The hours of instruction are thirty per week, and the duration of the session is about forty weeks.

The Day Apprentices' Course is intended for engineer apprentices and apprentice draughtsmen who are nominated by their employers. Students attend six hours on Mondays throughout the session.

In the Evening Course there are three sections: Preparatory, Introductory, and Mechanical Engineering course proper.

The Head of the Mechanical Engineering Department is Professor J. H. Smith, D.Sc.

## THE BELFAST ROPEWORK CO., CONNSWATER, BELFAST.

These works were established in 1876, being situated in close proximity to the shipbuilding yards and on the banks of the River Connswater, and at the present time the premises cover an area of about 35 acres.

The raw material used here comes from Russia, Italy, India, the Philippine Islands, and New Zealand. After being sorted, the manila fibres are taken to the hackling room, where the process is similar to that employed in the preparation of flax.

Hand roughing and machine hackling are both used. After hackling, the hemp is taken to the drawing and roving frames, where it undergoes a slight twisting, and the bobbins of roving are taken to the spinning machinery.

Both the wet and dry spinning processes are used, the former principally for fine yarns. The yarn then undergoes tarring, and when thoroughly dried is taken to the rope-walk to be twisted into strands, which are fastened to the hooks of machines called "travellers." These recede on tram lines from a stationary "fore-turn" or twisting head. Cords made from harsh or extra fibrous material have to undergo a scouring process prior to being sized and polished. The machines take from twenty to thirty separate twines at a time, which have been twisted upon bobbins, and are placed in front of the machine. After passing through the sizing troughs, the cords pass between two rollers by which the superfluous sizing is squeezed out. Thence they go over a large steam-heated cylinder, whereby a gloss is imparted to them, and they are then made up into balls. Thousands of different sizes and descriptions of ropes, lines, and twines are made, including binder-twine for harvesting purposes, fishing lines, plough lines, sash cords, etc.

The number of workpeople employed amounts to about 3,500.

## BROOKFIELD LINEN CO.,
## CRUMLIN ROAD, BELFAST.

The Spinning Mill of this company is situated in Crumlin Road, and contains 24,000 spindles, preparing machinery, etc., for flax and tow spinning.

The Brookfield Factory in Cambrai Street and the Agnes Street Factory contain 1,400 looms, with the necessary preparing machinery for weaving linen, union, and cotton goods.

The company also owns extensive bleaching and dye works.

The total number of workpeople employed is about 2,400.

## MESSRS. CANTRELL AND COCHRANE, AERATED AND MINERAL WATER MANUFACTORY, BELFAST.

The works of this firm are situated in Victoria Square, and were established by Dr. Cantrell in 1852 in Bank Lane, Belfast. In 1868 he was joined in Dublin by Alderman Sir Henry Cochrane, Bart., where a manufactory—the Nassau Works—was established. The firm at the present time has also depots in London and Glasgow.

The works in Victoria Square stand on what used to be the site of the Town Hall, and portions of this building now serve as counting-house and stores. A well was sunk at a cost of £2,000, and a plentiful supply of water from the Cromac spring was obtained at a depth of 116 feet. Latterly a second well has been sunk to a depth of 400 feet through red freestone. The supply is practically unlimited, about 18,000 gallons a day being pumped into an enormous slate cistern at the top of the premises. Before reaching this tank—though the strata through which it passes form an admirable filter—the water undergoes a final process of purification by being passed through a bed of sand and charcoal.

The charging of Cromac waters with gas generated from the action of sulphuric acid upon carbonates is the first stage after the pure water has been stored in the slate tank, which is covered with a glass lid. The gas is repeatedly washed, and is stored in gas-holders, whence it is drawn as required, a pressure of about 150 lb. per square inch being used to impregnate the water. The syrups for the various products are prepared in slate, porcelain, or glass vessels, in order to prevent the formation of injurious chemical deposits.

The bottling-room contains a number of continuous double-action soda-water machines, driven by overhead belting. Connected with them is a slate cistern and iced water, and everything is so arranged

that the processes of gas generation, charging, and bottling are continuous. To this room the bottles are conveyed down an inclined plane from the room in which they have been previously cleansed.

There are two systems of bottling—by hand and by machine. In the hand method the bottler sits in front of a machine; the bottle is placed beneath the mouth of the feeding-tube; the cork is inserted and, by means of a descending rod, is pressed nearly home; and the requisite quantity of syrup is forced into the bottle by the aerated water. The atmospheric air is next exhausted, the bottle quite filled, the cork driven home and wired or crown corked.

In another part of the factory the bottling is effected by means of automatic rotary bottling-machines. The machine is fed by one man with bottles and corks, the air being exhausted from the bottles as the apparatus revolves. During the revolution the bottles are syruped, filled with aerated water, corked, and delivered upon an endless band, whence boys pick them off and wire them. By the hand method one man generally bottled twenty dozen per hour, but by machine six times that number can be accomplished. In full-working summer days about 60,000 bottles are turned out per day. For sending abroad the bottles are doubly wired, wrapped in paper, and packed in barrels of straw. Amongst the various mineral and aerated waters produced at the Belfast and Dublin factories are soda, lithia, seltzer, and potass waters, ginger ale, lemonade, sparkling Montserrat lime-juice, etc. The number of men and boys employed in Belfast and at their Dublin factory is 500.

## MESSRS. DUNVILLE AND CO., ROYAL IRISH DISTILLERIES, BELFAST.

Although this business dates back to 1808, the present distillery was not erected till 1869, at which time the site it occupies as well as the surrounding district were green fields, and the part now covered by the bonded warehouses was a sheet of water, in some

parts 15 feet deep. The distillery comprises several blocks of buildings, built principally of perforated brick.

Facing the entrance are four Lancashire steam-boilers, each 30 feet long by $7\frac{1}{2}$ feet diameter, the flues being fitted with Galloway tubes. They supply steam to the engines and for the various requirements of the distillery, including the heating of the water used for the mashing. In the main flue is placed one of Green's economizers through which the feed-water is supplied to the boilers at a temperature ranging from 250° to 315° F. The higher heat is obtained in the earlier part of the week, when the economizer has the benefit of the waste heat from the pot-stills as well as from the boilers.

Behind the boiler-house are the engine-house and grinding mill. A vertical engine of marine type of 350 i.h.p., made by Messrs. Victor Coates and Co., supplies the principal motive power. There is also a horizontal engine in connection with the still-house, as well as several combined engines and pumps for feeding the boilers and for pumping the spent wash in the feeding-stuffs department. The grinding mill contains five pairs of millstones, each $4\frac{1}{2}$ feet diameter, driven at a speed of 140 revolutions per minute, and kept running day and night in order to supply the quantity of grist required for mashing. From the mill the grist is conveyed by an elevator and screw to the grist room.

On the left of the entrance is a huge grain silo which was erected a few years ago. This building consists of fifty-four bins with a storage capacity of 6,000 tons of grain. The grain is first of all discharged into two large hoppers, one at each end of the building, from which by means of band-conveyers it passes to large elevators travelling at a very high speed and which are capable of lifting 30 cwt. per minute to the top of the building, which is about 100 feet high. The barley is then screened, being afterwards distributed by means of electrically-driven bands to the pneumatic maltings, mill and kilns for germination, grinding and drying, etc.

The barley, after being thoroughly screened, is steeped for the usual length of time in conical steeping-tanks, of which there are six fixed above ten germinating drums. One tank can be made

to serve two or three drums. From the steeping-tanks the wet barley is run into the drums, and germination then takes place. To maintain a suitable temperature during germination a current of air is drawn through each drum. This air is cooled, moistened, and purified in an outside brick chamber before it reaches the growing malt, and after passing through the drum is discharged outside the building. The amount of air passing through the grain is regulated and controlled by a valve fixed at the outlet end of each drum. To turn the grain, the drums are revolved as often as the maltster may consider necessary.

The germinating process being completed, withering of the green malt takes place in the same cylinder, and is carried out on exactly the same principle as on the old floor system. When the operation is complete, the malt is discharged from the germinating drums to a band-conveyer and carried to an elevator, which discharges the grain into other drums where the drying and curing operations are conducted. It might be mentioned that under this system the grain is not damaged, as under the old floor system by the shovel or boots of the workmen.

The drying plant consists of four drums placed in two rows, one set above the other, the two upper drums for drying the malt and the two lower ones for curing it. Green malt from the germinating drums is filled into the drying drums by means of a band-conveyer and elevator, and hot gases are drawn through the malt from a small furnace by means of an exhaust-fan. The drying drums are kept revolving till the malt is in a fit condition to be dropped into one of the curing drums. In the curing drum the temperature of the malt is allowed to rise steadily, and the finishing-off temperature has to be maintained for some hours. The fuel generally used in the drying process is anthracite malting coal. The maltings are driven by a Tangye gas-engine of 80 h.p. worked from producer-gas.

Beside the maltings there is a large bonded warehouse fitted with an electrically-driven hoist. Beyond this building is the old malt-house, four storeys high, the roof of which is a wrought-iron tank about 112 feet long by 85 feet wide. This tank has a capacity of over a quarter of a million gallons, and the main object of its

erection was to ensure a good supply of water in case of fire. Next to this building is the mash-house, containing three mash-tuns, of which the largest is 29 feet diameter by 8 feet deep, having a capacity of over 30,000 gallons. Grist is passed into the various mash-tuns through Steel's mashers. All the pumping in this house is done by a rotary pump, driven at 500 revolutions, which raises 1,400 gallons per minute to a height of 50 feet. The cooling of the wort is effected by four large refrigerators. Close by is the tun room, containing sixteen fermenting wash-backs ranging in size from 27,000 to 35,000 gallons.

The still-house, situated to the right of the main entrance and near the engine- and boiler-house, is in communication with the mash-house. Here there are three pot-stills, holding together about 30,000 gallons, made by Miller of Glasgow, with the necessary adjuncts of wash-chargers and receivers for the low wines, feints, and whisky; also the glass safes through which the various distillates pass. From the still-house the finished whisky is run into vats in the adjoining spirit store, where the final operation of filling into the sherry and other casks takes place.

The various stages in the manufacture of Dunville's whisky, briefly stated, are as follows:—

(1) The barley is thoroughly cleansed by fanning.

(2) The barley is put into huge drums, heated to a temperature of about 100° F. to dry.

(3) The barley is then thoroughly soaked or steeped in immense vessels, the water being changed once or twice.

(4) The soft and swollen grain is then carried by pneumatic force to huge drums, where the process of germination or sprouting takes place. Germination converts the starch of the grain into sugar. This sugar nourishes the sprout or rootlet.

(5) Just when the sugar is most abundant in the sprouting barley the mass is transferred to drying chambers, where applied heat arrests the germination and clears the grain of its moisture. The grain at this stage is called "malt."

(6) The matured malt is passed to a mill, where it is crushed, forming a "grist."

(7) The grist is then mixed with warm water in a cylindrical vessel known as a "masher." The water extracts the saccharine matter contained in the grist or malt.

(8) The grist and water are conducted to a "mash-tun," a cylindrical vessel having a false perforated bottom, and fitted with revolving arms which mash the grist still further.

(9) The saccharified liquor is then drawn off, and is known as "worts." The residue—a moist meal—is used as a cattle or hog food, and is a valuable by-product of the business of distilling.

(10) The wort is run off from the mash-tun through the perforated false bottom into what is known as the "underback." After being cooled it is passed into the fermenting vessels; yeast is added, and the process of fermentation begins. The fermentable sugar by the influence of yeast is converted into alcohol and carbonic-acid gas. The fermented wort, known as "wash," is a fluid containing varying proportions of alcohol, unfermentable grain extract and water, and the object of distillation is to isolate the spirit as effectually as possible. This is done by distillation; that is, by converting the volatile constituents of the wash into vapour. The boiling-point of wash depends on the proportion of spirit which it contains. Alcohol boils at 173° F.

(11) The wash is then passed into Still No. 1 to be boiled or vaporized. The alcoholic vapour is condensed by cooling, and passes into a vessel known as a "receiver," and becomes known as "low wines."

(12) The low wines are then passed into Still No. 2 and vaporized, and condensed and carried into another receiver. This second condensation of the alcohol is known as "feints."

(13) The feints are passed into Still No. 3 to be vaporized and condensed. This third distillation produces whisky.

The principal warehouses for storing and maturing the whisky are situated about a mile from the distillery, in Adelaide and Alfred Streets; and there are also three warehouses adjoining the distillery, which hold about 10,000 butts. The total floor space of the warehouses is over 260,000 square feet, or nearly 6 acres. Between the distilleries and the warehouses a government staff of

two supervisors and twenty officers are constantly employed. About 40,000 gallons of whisky can be produced weekly, and a very large sum is paid annually in excise duty. A large proportion of the trade is done under bond; that is, the whisky is sent—without duty being paid in Belfast—to the various Crown warehouses all over the kingdom, where the excise duty is ultimately paid.

## MESSRS. GUNNING AND CAMPBELLS, FLAX SPINNING MILL, BELFAST.

This is an old-established flax spinning mill, containing about 16,600 spindles, and is situated at North Howard Street, Falls Road. It is driven by a set of triple-expansion engines, made by Messrs. Willans and Robinson, indicating about 400 h.p. and working with steam at 200 lb. pressure per square inch supplied by Lancashire boilers. The latter were specially constructed from the designs of Mr. A. Basil Wilson. The number of workpeople employed is 500.

## MESSRS. HARLAND AND WOLFF, QUEEN'S ISLAND SHIPBUILDING WORKS, BELFAST.

These works were started by Mr. (afterwards Sir) Edward J. Harland in 1859, and their area was about $3\frac{3}{4}$ acres. To-day they spread over more than 80 acres. In 1859 they employed 44 men, and now they find work for 16,000. Shortly after starting, Mr. Harland was joined by Mr. G. W. Wolff, and these two worked together alone until 1874, when Mr. W. J. Pirrie and Mr.

Walter H. Wilson were taken into partnership. In 1895 Sir Edward Harland died, and in 1904 Mr. Walter H. Wilson also died. Mr. Wolff retired a few years ago, and Lord Pirrie, as he became in 1906, now alone remains of those above-named, and he is chairman of the company.

In addition to the works in Belfast, they have works at Southampton, where 2,500 men are employed; they have recently acquired works at Govan, Glasgow, for shipbuilding and engineering, and are erecting works at Liverpool.

The first S.S. "Oceanic," the pioneer of the White Star Line, built in 1870, was the first vessel constructed with saloon and cabins amidships. This vessel, and the "Britannic" and "Germanic"—built for the same line—ran for over a quarter of a century with phenomenal success. The most notable productions during recent years are the following ships:—

*White Star Line.*—"Oceanic," "Celtic," "Cedric," "Baltic," "Adriatic," and "Olympic" (45,300 tons).

*Hamburg-American Line.*—"Amerika" (22,724 tons), "President Lincoln," and "President Grant."

*Holland-America Line.*—"Nieuw Amsterdam," and "Rotterdam" (23,980 tons).

*Red Star Line.*—"Lapland" (about 18,500 tons).

*Royal Mail Steam Packet Co.*—"Aragon," "Amazon," "Avon," "Asturias," and "Arlanza" (15,000 tons).

The work at present on hand includes large steamers for the White Star Line, Royal Mail Steam Packet Co., Bibby Line, Holland-America Line, Elder Dempster and Co., African Co., British and African Steam Navigation Co., and George Thompson and Co., Ltd.

The latest vessel built by them for the Royal Navy is the Admiralty yacht "Enchantress," and they have recently constructed the machinery for some of the largest vessels in the British Navy, including the first-class battleships "Hannibal," "Queen," "King Edward VII," and "Hibernia," the first-class cruiser "Minotaur," 27,000 i.h.p., and H.M.S. "Neptune," 25,000 s.h.p.

The firm has large facilities and experience in extensive repair and reconstruction work, having performed many difficult feats in connection with such work; for instance, the reconstruction of the "Philadelphia" (*ex* "Paris"), also the "China," after these vessels had been on the rocks; the cutting in halves and lengthening large steamers, such as the "Scot" and "Auguste Victoria"; the construction at Belfast of a new fore part for the S.S. "Suevic" and joining together the two portions of the vessel in drydock at Southampton. The capacity for production is about 100,000 tons and 100,000 i.h.p. per annum. There are eight slips for large ships.

The highest tonnage output was as follows:—

| Year. | No. of Steamers. | Tonnage. | I.H.P. |
|---|---|---|---|
| 1903 | 8 | 110,463 | 100,400 |
| 1910 | 8 | 115,861 | 100,130 |
| 1911 | 10 | 118,209 | 96,916 |

One of the latest developments within the works has been the complete electrification of the plant, the electric generating station being one of the largest private stations in the British Isles. Recently, alterations have been carried out at the north end of the Yard, the large building slips being greatly extended so as to facilitate the construction of the largest vessels, approaching 1,000 feet in length, and a massive structure has been erected over the slips with electric cranes, etc., for handling weights at any part of the building berths during the construction of the vessels. They have also a 200-ton revolving floating crane for putting engines, boilers, etc., on board vessels.

The works are well situated, being built on what is called Queen's Island, on one side of them being the Victoria Channel, and on the other the Musgrave Channel. Both of these unite in the New Channel, which has direct access to Belfast Lough, and thence to the sea.

The work of the offices is divided into five main departments, which have their sub-divisions, each sub-division having its own room. The office staff numbers between 400 and 500. For practical purposes the works may be considered as being divided into two separate parts—shipbuilding and engineering.

*Joiners' Shop.*—This shop is divided into two bays with a gallery. On the ground floor the tools comprise saws, planers, drills, moulding machines, etc., and the gallery contains similar tools. All sorts of wood constructional work in connection with shipbuilding is carried out in this department, including doors, partitions, etc.

*Beam Shop.*—Adjoining the mast shed is the beam shop, where beams, rails, plates, angles, cement, etc., are stored.

*Mould Loft.*—This loft, situated above the plumbers' shop, has a splendid floor where the lines of the vessels are laid out.

*Plumbers' Shop.*—The machinery here is all electrically driven, and comprises cold saws, punching, shearing, screwing, drilling, and bending machines, lathes, and jib-cranes. Narrow-gauge lines run in different directions in the shop.

*Platers' Sheds.*—There are several of these sheds, well equipped with the usual tools, among which may be noticed an angle-bending machine by Hugh Smith and Co., Glasgow, a large hydraulic shearing machine by J. Cameron, Manchester, and a right-angled plate-planing machine by Hetheringtons, of Manchester. The North Platers' Shop is intended to supply material for the vessels which are built on slips 1, 2, and 3. The West Platers' Shed is intended to provide for the wants of slips 5 to 7, and belongs to the older part of the establishment.

*Gantries.*—The gantries connected with the No. 1 slip run on twelve wheels on each side, there being a double line of rails, $6\frac{3}{4}$ inches wide, placed at 100 feet centres on each side of the slip.

The latest gantry was constructed by Sir William Arrol and Co., the electric crane equipment having been supplied by Stothert and Pitt, Bath. The area covered exceeds 840 feet long by 270 feet wide, and the diameter of the circle covered by the largest crane is 270 feet, while the highest point from ground level is 228 feet.

Three long lines of tall steel lattice-towers run parallel to one another at a distance of over 120 feet between the lines. Each line consists of eleven towers placed at 80-foot centres. Along the road of the centre line of towers runs a Titan crane, which has a reach of 135 feet, and can deal with a 3-ton load at this radius. Ships practically 1,000 feet long can therefore be dealt with.

*Turning Shop.*—Various tools by Shanks, Hulse and Co. and other well-known makers are installed in this shop. In this bay there are two overhead electric travelling-cranes each capable of lifting 20 tons.

*Engine Erecting-Shop.*—In this bay there is one pit for erecting engines. The majority of the tools here are for dealing with heavy work. There are four overhead electric travelling-cranes of various powers, the largest being capable of lifting 40 tons.

*Engine Fitting-Shop.*—The tools in this shop are not quite so heavy as in the previous shop. The tool equipment comprises milling-machines, planers, drills, etc. White-metal bearings for large engines are prepared in this department. The white metal after being run is carefully consolidated by being hammered all over with a round-ended hammer before being bored. The hydraulic testing of valves and small cylinders is also carried out here. The shop is served by two overhead electric travelling-cranes, each capable of lifting 10 tons.

*New Erecting-Shop.*—This bay is 725 feet 6 inches between centres of end columns and 80 feet between faces of columns. The roof is entirely covered with glass. A special erecting bed has been made for erecting the heaviest class of reciprocating engines. Turbine machinery is also assembled in this bay. At the east end are large tools by Shanks and Co., for dealing with the heaviest class of work. There are also two vertical cylinder boring-machines. This bay is served by three overhead electric travelling-cranes, two capable of lifting 80 tons each and one capable of lifting 20 tons, the 20-ton crane running in gantry at lower level than the 80-ton cranes.

*Brass-Finishing Shop.*—Here every kind of brass article required

in shipbuilding and engine work is machined and finished, all the tools being driven by belts. The main shafting is electrically driven.

*Smiths' Shop and Forge.*—This is divided into two bays. There are altogether sixty smiths' fires. The blast is provided by three blowers driven by electric motors. At the south end are two forge furnaces; the largest hammer is 30 cwt. All hammers are driven by pneumatic pressure.

*Pattern Shop and Store.*—This building stands apart. The shop is equipped with circular-saws, planers, band-saws, emery-wheels, lathes, universal pattern-making machines, etc. Suction arrangements are provided for withdrawing the sawdust from the machines.

*Iron Foundry.*—This is divided into six bays, and is served by six cupolas having a large capacity. Pig-iron, scrap, and flux are taken up to the charging platform by overhead electric travelling-cranes; four of the cupolas are arranged in a group, so that they can all be charged from the same platform; two cupolas are grouped together and served from a separate platform.

*Fettling Shop.*—Pneumatic tools are largely used in this shop, and a number of Tabor moulding-machines of various sizes are in use.

*Brass Foundry.*—"Brass" in this case includes not only brass, but all sorts of bronzes, gun-metals, white metal for bearings, etc. Propeller-blades and other heavy articles are cast here. For melting the various metals there are pot and reverberatory furnaces. Two tilting furnaces are placed at one end of the building; Tabor moulding-machines are largely used.

*Boiler Shop.*—A great variety of work is carried out here in addition to actual boiler-making, such as the construction of funnels for vessels, which now attain enormous dimensions. On entering the shop one notices a right-angled plate-edge planing-machine by Hetherington and Co., which will take plates up to 36 feet long. Two 20-h.p. 440-volt Vickers motors are geared to this machine. There are also a large vertical boring and surfacing machine by Embleton and Co., and two tube-plate cutting-machines. Of the hydraulic machines there are four riveting-machines and

two flanging-machines. Another large tool is an upright hydraulic plate-bending machine, which will take in plates up to 12 feet wide. Other tools in this shop are radial drills, punching and shearing machines, straightening rolls, etc. This shop is served by overhead electric travelling-cranes of various powers, the two largest being capable of lifting 50 tons each.

*Spar-making Shed.*—The raw timber is adzed, and then smoothed and polished. Although steel masts and spars are being increasingly used, there is still some demand for wood in this direction. In a loft above this department the sails and awnings are made, and on the same level is the upholstering department, which is of a most extensive character.

*Timber Sheds.*—The sheds for the drying of timber, of which the value of the stock is about £250,000, cover an area of about 80,000 square feet, and the sides are open. Before being stored the logs are sawn up into approximately the sizes required later, and are then stacked horizontally with wedges between each board or plank.

*Central Power-Station.*—This contains a well-arranged plant, and, if necessary, it can supply current to light continuously 133,000 incandescent lamps (8 c.p.) taking 30 watts each. The whole of the building and plant was designed and installed by the firm's own engineers. A dual plant has been installed; both alternating and direct-current machinery to generate electricity at about 450 volts has been laid down. The building is 415 feet long by 68 feet wide, and there is an annexe containing some electrically-driven hydraulic and air-compressing machinery.

The main boiler equipment consists of five single-ended marine boilers, manufactured by the firm, besides two Babcock and Wilcox and one Yarrow marine-type water-tube boiler, having a total heating surface of 18,781 square feet. One of these boilers is heated by the gases produced from the burning in a Meldrum furnace of the refuse collected in the works, which sometimes amounts to 35 tons per day. The water for the boilers is passed through a Paterson feed-water purifier.

The engine-house is 171·6 feet long by 64 feet wide. There are

four main generating sets, each being driven by a four-cylinder twin tandem triple-expansion horizontal engine of the drop-valve type, made by Sulzer and Co., Winterthur. Each engine has two cranks driving on to one crankshaft. The generators were made at Frankfort by the Lahmeyer Co. The sets 3 and 4 are identical and differ from Nos. 2 and 5 in the fact that two generators, one direct-current and one three-phase, are carried on the shaft driven by the engine. The other equipment of the engine-house is very complete. Set No. 6 consists of a three-cylinder vertical compound engine, coupled direct on the same bed-plate to a 350 kw. three-phase alternator. No. 1 set consists of a compound three-cylinder vertical enclosed engine driving a 60 kw. balancer, to which a 150 kw. 440 V.C.C. generator is also coupled. For the field excitation of the large generators, etc., two 60 kw. enclosed sets are run in parallel with the battery; these high-speed sets were manufactured by W. H. Allen, Son and Co., Bedford. Arranged under the switchboard gallery opposite to the entrance are battery boosters, motor generators, balancers, etc., and running along the engine-house is a 15-ton electric travelling-crane.

Next to the boiler-house is a battery room, containing an accumulator with a capacity of 1,000 ampere-hours. Above this are placed some oil-tanks and two water-reservoir tanks, capable of holding 35,000 gallons, whilst adjacent thereto are an electrical test-house, stores, and a workshop.

The hydraulic machinery consists of four electrical-driven pumps capable of giving an aggregate of 500 gallons per minute at 800 lb. per square inch, and 260 gallons at 1,500 lb. per square inch.

The air-compressors, also electrical-driven plant, consist of three sets capable of jointly giving 5,500 cubic feet of air at 100 lb. per square inch.

*Alexandra Dock Works.*—These are largely devoted to rough wood-working, and to the storage of baulk timber, iron, oil, etc. There is also an oil-fuel storage-tank with pump, the former being isolated and surrounded by water. In another building may be seen samples of various styles of decorations for saloons, state rooms, and cabins.

# MESSRS. INGLIS AND CO., BREAD AND CAKE MANUFACTURERS, BELFAST.

This company was founded by Mr. James Inglis, who went to Belfast in 1871, at the time when the transition from bread being baked in the home to that produced in factories was taking place. Within a short period the public demand had increased to such an extent that more commodious premises had to be secured. In 1882 the present site was chosen of one of the largest single bakeries in the world. Centrally situated, convenient to the port and principal railways, the factory is bounded by Eliza, McAuley, Stewart, and Welch Streets, covering an area of two acres, on which have been arranged a series of substantial stores and bakehouses, specially designed and constructed for the production of bread under the most hygienic, scientific, and economic conditions.

The bakehouses contain over 60 great ovens and three automatic baking plants, and when taxed to their utmost capacity are capable of producing, during one shift of eight hours, 40,000 loaves weighing 40 tons, so that within a single day 120 tons of loaves can be baked and distributed throughout Ulster. Six motor vans and 220 horses are employed in this distribution.

The power and light required to serve the stores and bakehouses are obtained from a double set of Diesel oil-engines, by means of which electricity is generated and conveyed throughout the factory.

The Inglis Employees' Benefit Trust was formed in 1910, and is administered by a council composed of three nominees of the directors of the company and three nominees of the employees. The amount to the credit of the fund is over £3,000, and the interest on this, together with allocations received from the company, is utilized to help employees or their families in time of sickness and trouble.

The number of employees is about 500.

## MESSRS. WORKMAN, CLARK AND CO., SHIPBUILDING AND ENGINEERING WORKS.

The original works of this firm, which was founded in 1880, covered only about 4 acres of land on the north side of the River Lagan, to the west of Belfast Harbour. Progress was rapid and continuous, new berths being added and new shops built and equipped, until in 1891 the shipbuilding yard covered an area of about 15 acres. In 1905 and 1911 further extensions were made to the machinery department, and now the total area covered is 829,724 square feet, of which 220,000 square feet is roofed in. In 1910 additions were made to the North shipyard, 12½ acres being added, bringing the total area of the firm's works to 82½ acres, while the number of berths is now twelve, capable of taking the largest ships that are likely to be built for some years.

The result of this development in the works is shown in the following statistics:—

*Output in Quinquennial Periods.*

| Number of Ships Launched. | Five Years' Total Tonnage. | Average per Year. | Five Years' Total Indicated Horse-Power. | Average per Year, Indicated Horse-Power. |
|---|---|---|---|---|
| | | Tons. | | |
| 1880–84 inclusive | 27,762 | 5,552 | — | — |
| 1885–89 ,, | 38,883 | 7,777 | — | — |
| 1890–94 ,, | 124,145 | 24,829 | 34,910* | 11,636 |
| 1895–99 ,, | 191,710 | 38,342 | 120,120 | 24,024 |
| 1900–04 ,, | 270,525 | 54,105 | 172,550 | 34,510 |
| 1905–09 ,, | 326,168 | 65,234 | 254,350 | 50,870 |
| 1910 (one year) | 49,993 | — | 36,300 | — |
| 1911 ,, | 66,399 | — | 51,800 | — |
| | 1,095,585 | — | 670,030 | — |

* Three years.

The work done has included nearly every type of merchant ship, from the high-speed liner to the tramp steamer. The more important lines which have had large additions to their fleets from the works are the Royal Mail Steam Packet Co., the Allan Line, the Peninsular and Oriental Line, Alfred Holt and Co.'s Blue Funnel Line, Messrs. Lamport and Holt, the Ellerman Lines, the British India Co., the Harrison Line, the Tyser Line, and the Cunard Co. Perhaps the most notable steamer built was the Allan Line turbine-steamer "Victorian," which was the first Atlantic liner to be fitted with the Parsons turbine. Reference should also be made to the large refrigerated meat-carrying ships built. One of the latest, the "Muritai," completed last year for the Tyser Line, had a capacity of 292,000 cubic feet in refrigerated rooms for carrying meat from South America to this country.

## The South Yard.

The South Yard, situated opposite the North Yard, was acquired by the firm in 1894, and largely rearranged; like the North Yard, it is complete within itself. There are five berths. These take ships ranging in length from 523 feet to 417 feet, and in beam from 61 feet to 53 feet. The ground was made up of dredgings from the river at an early stage in the development of the harbour, and consequently very heavy piling had to be done, not only in the berths, but for the columns of the roofs and bed-plates of the large machine-tools. Good foundations were obtained generally at 40 feet, and the piles were driven in lengths and scarfed at the junctions.

## Engineering Works.

The Engineering Works of the Company are situated at Queen's Road, opposite to the South Yard. As in the case of the shipbuilding works, there have been considerable extensions in recent years. The works had originally been laid out so that the various workshops could be extended in an easterly direction without any further interference with the general convenience of the plan, from the standpoint of economic handling of heavy loads and the

sequence of operations. The Boiler-Making Department, consisting of six bays, is situated on the left side of the main entrance, where are located the administration offices, drawing office, and other departments associated with the administration and the inspection and other work, while to the rear of them is the power-station for the Engine Works and the South Yard. The Engineering Department is under separate organization, and therefore the offices are extensive, there being a separate drawing-office of 56,300 square feet, well lighted from the roof, while the chief draughtsman has a separate office adjoining, and the Estimating Department contiguous to it.

*Power-Station.*—This is situated to the rear of the main offices, and has a total capacity of 1,500 kw. Steam is generated in two boilers of the marine cylindrical type; one of them is 15 feet 6 inches in diameter and 10 feet 6 inches long, and the other 14 feet 6 inches in diameter and 11 feet long, each with three furnaces. The boilers work under natural draught. There are two electric generating sets, one driven by a triple-expansion engine running at 95 revolutions, and developing 1,000 i.h.p., and this is coupled direct to a direct-current 600-kw. generator. The exhaust from this engine is passed to a Brush-Parsons turbine, working with an initial pressure of about 20 lb. absolute, and having four stages for expansion, the rotor being 14 feet 6 inches long and of 7 feet 6 inches diameter. This turbine is coupled direct to a generator of 600-kw. capacity when running at 2,000 revolutions per minute. Electricity is distributed by conductors of the armoured-cable type, carried overhead on timber columns. The motors range in power from 75 h.p. down to 10 h.p., excepting the motors driving air-compressors and hydraulic-pumps in the power-station.

Pneumatic power is extensively applied throughout the works, and air-compressing plant is installed in a building to the rear of the Boiler Shop. The air-compressors are of the two-stage type, driven by 100-h.p. electric motors at 250 revolutions per minute, the output of the compressors being 500 cubic feet of free air per minute at 100 lb. per square inch.

There is one set of hydraulic pumps in the engineering department, actuated by a 60-h.p. electric-motor; the plungers—three in number—being 4 inches in diameter. The pressure is 1,000 lb., and the power is used in flanging, riveting, and other machines, in hydraulic jib-cranes, and in the testing of boilers. The accumulator has a load of 55 tons. As in the case of several of the hydraulic tools in the shipyard already described, Messrs. Hugh Smith and Co.'s principle of economising power-water is utilized.

*Engine Department.*—The main engine department includes five bays, three of these being 240 feet long, and respectively 55 feet, 60 feet, and 50 feet spans. The two others, of 30 feet spans, were originally of the same length, but last year they were increased to 400 feet. No. 1 bay forms the Erecting Shop; No. 2 bay is utilized for the erection of reciprocating engines; and No. 3 bay is a Machine Shop. The practice in the Erecting Shops is to build the engines on the floor in Nos. 1 and 2 bays, the height of the crane track being 45 feet. The engines when completed are placed on trucks and passed to the ships over standard-gauge railway track, leaving the works through the doors in Queen's Road. In the Turbine Shop the most notable tool is a lathe for turning the rotors of turbines, and a boring-mill for the casings. In No. 2 bay there are several important tools. The largest of the wall planers has a vertical travel of 17 feet 6 inches, and planes a side surface of 21 feet, the table being 10 feet wide. The largest of the boring-mills for the vertical cylinders of marine engines takes jobs 12 feet high, the centres between the columns being 12 feet. The third bay is utilized for marking off, building up crankshafts, reamering out the couplings, drawing on the propeller shaft liners and fitting the propellers together.

*Boiler Shop.*—This shop is well equipped and contains six bays. No. 1 bay, of 30 feet span, is taken up partly by the marking-off tables, and the construction of uptakes and light work generally. No. 2 bay, of 50 feet span, served by a 25-ton overhead-traveller, is devoted to the making of combustion-chambers and the drilling and riveting of boiler-plates. No. 3 bay is largely utilized for the

assembly and riveting of boiler-shells. In bay No. 4, the drilling, tapping, and staying of the boilers and the construction of combustion-chambers are undertaken, as well as funnel-construction. The plates are marked off and flanging work done in No. 5 bay, and in No. 6 bay there is the smiths' shop, where light angle-iron work is done. The boiler-tool store is situated adjacent to this shop. As to the boiler-making machinery, mention may be made of the boiler-shell drilling-machine, a four-headed drill, having a bed 34 feet long. This machine takes in the largest boilers, and drills four strakes simultaneously, the motive power being a 35-h.p. electric machine. A notable tool is a double-column horizontal drilling, tapping, and staying-machine, of Messrs. Campbells and Hunter, Ltd. The machine drills and taps stay-holes 2 inches in diameter in the backs of marine boilers through both shell and combustion-chamber plates, and screws the stays in position. The machine is driven by a 15-h.p. motor at 700 revolutions per minute. The flanging-machine has been supplied by Messrs. Hugh Smith and Co., Possil Park, Glasgow. It is used for flanging combustion-chamber plates, one ram being used for holding down the work and the other for flanging the plate. Man-holes and furnace-mouths are also done on this machine.

In the Engineering Works there are various miscellaneous shops, some of them with notable equipment. In the Pattern Shop, which is 180 feet in length by 50 feet span, there is a good collection of machines, including lathes, circular and band-saws, and planing-machines, all of them belt-driven from a line-shaft, which is rotated at 750 revolutions per minute by a 20-horse-power motor. The Blacksmiths' Department is accommodated in a separate building, 180 feet in length, with two spans, one of 50 feet and the other of 25 feet 3 inches. There are thirty-two fires arranged as shown on the plan. These are of the open-hearth type, the blast being provided by a Sturtevant blower. The steam-hammers range from 20 cwt. down to 5 cwt., and there is in addition a 15-cwt. air-hammer.

Centrally situated between the Engineering Boiler-Shops is a general store, 70 feet in length by 39 feet 6 inches in width; but

independently there are at the end of the Smithy separate stores for wrought-iron bars, used for making gratings, a fire-clay and brick stores, and a separate room for cast steel and for making tools. The Plumbers' Shop is located at the eastern end of the site, and amongst other tools in it there is a 6-inch and 3-inch screwing-machine driven from a line-shaft by a 5-horse-power motor at 1,100 revolutions.

## The North Yard.

This yard, the one in which shipbuilding was first commenced by the firm, has seven shipbuilding berths. Two of these were built in 1910, when $12\frac{1}{2}$ acres were added to the area of the works, in order to arrange for berths to take ships of 1,000 feet in length. At present, however, the largest ship building is 560 feet long by 68 feet beam, for the Alfred Holt Line, the tonnage being 14,200 tons gross register.

*Platers' Sheds.*—These cover an area of 65,780 square feet, and are contiguous to berths 1 to 4. The largest plates hitherto dealt with have been 34 feet 8 inches long by 6 feet 6 inches wide and $1\frac{1}{8}$ inch in thickness, but the tools are capable of dealing with still larger plates. Of the three platers' sheds, the principal one is located between berths 3 and 4. The plate furnaces are 56 feet long and 7 feet 6 inches wide; they have six fire-boxes with double flues, and are coal-fired. The slabs on which the plates are worked to the required form are 25 feet long by 19 feet wide. The plate "mangles" are by Messrs. Craig and Donald. The bending rolls are fitted with two 15-h.p. series-wound motors, one at each end, with tramway controller for lowering and lifting the rolls, through belting, while the rolls are run by a 100-h.p. motor. There is a complete installation of punching and shearing machines, capable of punching $1\frac{1}{2}$-inch holes through $1\frac{1}{2}$-inch plates, the shears taking the same thickness of plates. The plate-edge planers take in plates 35 feet in length by 6 feet in width, the two sides being, of course, cut simultaneously. The other appliances are in accordance with the latest practice. In the shops there is a series of jib-cranes to

lift 5 tons, with sufficient radius to enable plates to be passed from one machine to the other without being handled by manual labour.

*Frame and Beam Shop.*—This is in two bays, one of which is 564 feet in length and the other 380 feet in length, the span of the Belfast-roof construction, in both cases, being 80 feet. At the further end of this bay there are two double-ended framed reverberatory furnaces 80 feet long by 7 feet 6 inches broad, having six fire-boxes with double flue. Thus the bottom and side frames of two vessels can proceed simultaneously. The slabs at each end of the furnaces are 61 feet by 38 feet. The largest angles hitherto worked are 6 inches by 6 inches by $\frac{18}{20}$ inch thick, and the largest channels 12 inches. The winches work through block and tackle. The firm were the first users of hydraulic frame-bending machines, the machine being the invention of Mr. Wm. Campbell, who was shipyard manager from the start of the firm until he retired five years ago. Contiguous to the bending-slabs are the scrieve-boards, which have an area the one of 72 feet by 80 feet and the other 80 feet by 80 feet. In this shop are four-sided punch and shearing-machines, specially arranged to punch $1\frac{1}{2}$-inch holes at two ends, 6-inch limber holes at a third side, shearing at the fourth side; also the largest size of Lambie type of hydraulic combined manhole punch and joggling machine, capable of joggling 12-inch channels and bulb-angles, besides hydraulic channel-cutting machines, hydraulic beam-bender and two large angle-cutting machines of Pels' type. A feature of the hydraulic machines in the yard is the adoption of Messrs. Hugh Smith and Co.'s system of power-water economiser, notably in the joggling-machines, the cutters for 12-inch channels and the beam-benders. It is claimed that this system saves about 60 per cent. of the hydraulic power-water because the high-pressure water is not used until it is actually required. The frames and beams when formed to the desired curvature are taken out to the frame skids.

*Joiners' Shop.*—This is arranged alongside the Milewater Wharf, and is a two-storey building, each floor having an area of 21,600 square feet. The lower floor is arranged for the machine-tools driven from overhead shafting actuated by an electric motor.

There is a complete variety of wood-working tools. On the upper floor are 108 benches. The store of timber in sheds arranged where convenient throughout the works is valued at £100,000.

*Smithy.*—This forms part of the building in which the heavy platers' machine-tools are situated, although divided from it by partitions. There are here a $12\frac{1}{2}$-cwt., two 10-cwt., one 7-cwt., two 5-cwt., and a $2\frac{1}{2}$-cwt. steam-hammer, along with a Massey hammer, principally used for drawing out packing-pieces for the shell-plating. There are 38 fires, and the production averages about 7 tons per week. Much of this smiths' work is passed to the mechanics' shop, where all the ship-work is machined, including valves, lifting-gear for skylights, cargo-blocks, and steering-gear, while at the same time extensive repair work is undertaken, and the flat keel-plates of ships are planed.

*Shipbuilding Berths.*—There are seven shipbuilding berths, and these are so disposed that two of them may accommodate any ship of the maximum length probable for many years. With one exception these berths take ships ranging in length from 500 feet upwards. On each side of each berth there are arranged electric derrick-cranes, of 4 tons lifting power, controlled from the bottom, so that the fitters can themselves work the cranes. The derricks are 150 feet high, and the jibs have an overreach of 50 feet. On each of the new berths, however, there are now being constructed three travelling cantilever cranes, of 8 tons lifting power, by Sir William Arrol and Co., Ltd., Glasgow. The motions of hoisting, slewing, traversing, and travelling are operated by independent motors, with continuous current of 230 voltage. The tower is square in section, and built up of rolled steel angles and plates.

*Launching Mechanism.*—The launching weight of the heaviest ship so far set afloat has been 8,050 tons, and the firm, as a rule, use in such case ways 3 feet 9 inches in width. The angle at which the berths are arranged gives adequate launching ground in the River Lagan. Recently the firm have had constructed a special launching gear, by Sir William Arrol and Co., Ltd. The gear is of the type in which a heavy tumbler-pawl, held in check by an hydraulic piston, is fixed in each of the ways and is used to hold

the vessel in check ready for the launch. At the required instant the pressure behind the hydraulic piston is relieved and the tumbler-pawl falls over, leaving the vessel free to slide down the ways.

The ships built at the North Yard are, for the most part, completed at the Milewater Wharf, where there is a 5-ton steam-crane with a radius of 50 feet, the jib having an 80-foot lift. Where heavier loads have to be put on board, the ship is removed to the wharf contiguous to the Engine Works, where there is a 100-ton crane. At the present time there is being constructed by the Belfast Harbour Commissioners a dolphin between Nos. 5 and 6 berths, and here vessels up to 1,000 feet long can be accommodated, while alongside the eastern boundary, and therefore adjacent to the extension works, berthage can, if necessary, be constructed.

*Power Installation.*—The power-station, situated alongside the Milewater Wharf, contains two generating sets. There is a triple-expansion Bellis and Morcom engine, to each end of which is direct-coupled a continuous-current generator of 300 kw. capacity. This engine exhausts to a Brush-Parsons low-pressure turbine which drives a generator of 600 kw. capacity. A third engine, for night duty, is by Messrs. Victor Coates and Co., Belfast, and drives a generator of 150 kw. capacity. Compressed air is used throughout the works, and there are three compressor sets in the power-station producing air of 100-lb. pressure. The hydraulic power installation is of interest by reason of the somewhat unusual arrangement of the accumulators, which are three in number. The pumps are electrically-driven, and there are three sets with plungers $2\frac{1}{8}$ inches, $3\frac{1}{2}$ inches, and 4 inches in diameter respectively. These are driven through belting and work at a pressure of 1,350 lb.

[*This Notice was prepared at the request of the Firm, from the illustrated Article in* "Engineering," 19*th July* 1912, *by permission of the Editors.*]

## YORK STREET FLAX SPINNING CO., BELFAST.

These mills, situated in almost the centre of the city, were founded in 1830 by the late Mr. Andrew Mulholland, and the Mulholland family, whose head is Lord Dunleath, have retained an interest in the business since it was transformed, in 1864, into the company with its present name. In addition to the buildings in York Street, another large mill for spinning flax yarns and sewing threads has been purchased in a neighbouring district of the town, and the company are thus able to carry on all the manufacturing operations required for converting the raw flax into finished linen.

The works occupy an area of 786 feet length by 221 feet width, or about 4 acres. The central part of the west end, a fire-proof building of eight storeys, occupies a space of 124 feet length by 50 feet width. In it are stored flax, tow, dressed line, brown cloth, etc.

Flax is the product chiefly of four countries—Ireland, Belgium, Holland and Russia. It has very different characteristics, according to the locality from which it comes. Before the flax reaches the spinner it has undergone a preliminary process known as "retting," which consists of steeping in water the bundles of flax that have been pulled out of the ground. The object of the retting is to decompose by fermentation the gum which holds the straw and fibre together. After the retting has been carried far enough the flax is dried and taken to the Scutch Mill, where it is roughly cleaned to free the fibre as much as possible from the straw. After this process it passes into the hands of the spinner, and it is, after being thus retted and scutched, that the flax is seen in the Flax Store.

The south wing has five storeys, and contains hydraulic presses, crane pumps, and gas-engine to drive them. The north wing, 229 feet by 46 feet, has six storeys, and forms the Preparing Mill,

In this department are various machines working in sets of four or five. The first machine of the series is called the "Spread Board," and here the flax is laid in wisps on travelling bands in long continuous lines, and is then passed through sets of rollers which draw one fibre away from the other, and produce a thick ribbon containing millions of separate fibres. As these ribbons emerge from one machine they are passed to another, and at each step they are drawn out longer and thinner, until in the last of the series, called the "Roving Frame," the fibre is twisted into a loose, thick thread, and then wound on large wooden bobbins.

The next stage is the spinning. Before it reaches the spindles, of which there are 63,000 in the Company's two mills, the thread passes through a trough of hot water. The twist or "spin" converts the fibres into solid round threads, which are known as "yarn." It is now, so far as processes of manufacture are concerned, practically a finished article of commerce, except for some minor processes of drying and winding. Before it reaches the loom it has frequently to undergo considerable changes, the principal of which are "boiling" and "bleaching." Boiling reduces the bulk of the yarn, and enables it to be woven into a tight and firm cloth. Yarns which are bleached are mostly such as are intended for glass cloths, towels, etc.

At the south-east corner of the block is a five-storey building, in which the first floor is used for weaving, the second for pirn winding, the third for yarn dressing and beaming, the fourth for hank winding, and the fifth floor for dressing. The weaving sheds contain about 1,000 looms for plain and damask linens. Bleaching, dyeing, and finishing of linens and yarns are carried on at Muckamore, about 20 miles from Belfast.

The steam plant consists of eight Lancashire boilers and three beam-engines with Corliss valves. One engine, with 35-inch cylinder and 5 feet stroke, making 45 revolutions a minute, drives the Preparing Mill. The two others, with 35-inch cylinders and 7 feet stroke, making 32 revolutions a minute, drive the Spinning Mill. For driving the Weaving Factory and heavy finishing machinery there are four Lancashire boilers supplying steam to

two beam-engines, with 38-inch cylinders and 7 feet stroke, which have wrought-iron beams and Corliss valves, and make twenty-nine revolutions per minute. They drive two main shafts direct from the fly-wheel.

The manufactures include fronting linens of different qualities, interlinings, printed linen shirtings, dress linens and lawns, etc., towels, hollands, handkerchiefs, sheetings, damasks, etc., and various goods for the West Indies and Spanish colonies, such as creas, platillas, bretanas, silesias, irlandas, etc. Besides 4,500 regular workpeople employed, large numbers are also engaged in the country in embroidery and fancy work, and also on the bleach-green.

## GIANT'S CAUSEWAY ELECTRIC TRAMWAY, PORTRUSH.

This tramway is the premier electric tramway. It was opened in 1883, introducing for the first time electricity as the traction power—at least in a practical form beyond short experimental lines—and working it as a hydro-electric tramway by water-power on the River Bush. It has also many other unique features; it was the first tramway or light railway constructed along the side of the public roadway and upon a pathway, specially constructed, raised above the surface of the roadway. It was first constructed with a side electric conductor-rail—third-rail system—using only a current of 250 volts; but in 1898 it was altered into the overhead trolley system, using a current of 550 volts. The Managing Director and Engineer is Mr. William A. Traill, M.A.Ing., who was the constructor of the entire tramway and works. The late Sir William Siemens designed the first electric equipment.

# MEMOIRS.

George Henry Allen was born in London on 7th July 1891. He was educated at Christ Church School and Sir Walter St. John's School, Battersea, from 1899 to 1905, and entered the drawing office of Messrs. Royce, Ltd., Trafford Park, Manchester, for one year. This was followed by his spending a year in the drawing office of Messrs. Tylor and Sons, York Road, London, and during the evenings he attended classes in Applied Mechanics and Mathematics at the Battersea Polytechnic. In 1908 he worked in the shops and drawing office of the Thames Ironworks and Engineering Co., Greenwich, and in June of the following year he went to the Acton Vale works of Messrs. D. Napier and Sons, where he was employed as jig and tool draughtsman. His death occurred at the age of twenty-one in endeavouring to save the life of another. In August 1912 he was staying at Rottingdean, near Brighton, with a party of young men who were members of the Caius College Mission Church, Battersea. While bathing on Sunday, 4th August, one of the party was seized with cramp and carried out to sea and drowned; three others, including Mr. Allen, went to his assistance, and in their attempt to reach him they were also drowned. He became a Graduate of this Institution in 1910.

Edwin Ault was born at Cannock, Staffordshire, on 17th July 1848. He served his apprenticeship with Mr. Isaac Shone, civil and mining engineer, from 1863 to 1869. On its completion he acted as engineer to the Talk o' th' Hill Colliery to 1870, and then as engineer to the Brymbo Colliery to 1873. He then returned to Mr. Shone as his chief assistant until the end of 1882, when he joined him as partner, and assisted in the designing and construction of ejectors, air-compressors, etc. His death took place at Lahore from heat stroke, on 24th June 1912, in his sixty-fourth year. He became a Member of this Institution in 1892.

George Bird was born in London on 19th January 1863. After being educated privately he became a student at the Crystal Palace School of Engineering for one year, 1879–80, and then was apprenticed to Messrs. Turner and Co., general engineers, Manchester Square, London. In 1882 he went as assistant to Mr. E. Perrett, of Westminster, until 1886, during which time he was engaged in experimental work on superheated steam-engines, steam-tram engines, etc. In 1887 he became a partner in the firm of James Bartle and Co., engineers and ironfounders, Lancaster Road, London, W. In this connection he remained until his death, which took place on 7th June 1912, at the age of forty-nine. He became a Member of this Institution in 1891.

James Dunn was born on 6th March 1838. At fourteen years of age, after having been trained in a private school and technical classes at Chatham, he entered upon his apprenticeship at the Chatham Dockyard. From this time onwards, covering a period of almost sixty years, he has taken a part of progressive responsibility in the development of the armoured warship. On the completion of his apprenticeship he spent a year in the drawing office at Chatham, and then he was promoted in 1860 to a draughtsmanship in the Constructive Department of the Admiralty, continuing in this position until 1867, when he went to the Clyde as an overseer of one of the earliest of the ironclads built by contract. In 1869 he became chief draughtsman at the Admiralty; in 1874, assistant constructor; in 1879, constructor; in 1880, a member of the Royal Corps of Naval Constructors; and in 1894, Senior Constructor and Assistant Director of Naval Construction. In addition to the strenuous work he carried out, his services were frequently called upon in connection with many Government Inquiries. In 1875 he was requested by the Admiralty to survey many merchant ships, with a view to their selection to act as cruisers in time of war. In 1884 he was requisitioned by the War Office to supervise the preparation of the vessels to proceed to the relief of Gordon at Khartoum. He acted as the naval construction adviser to the War Office throughout the whole period, and three successive Secretaries

of State for War especially thanked him for his services. In 1885, when the first Load-Line Committee was appointed, he was again the representative of the Admiralty, and thirteen years later, when a further committee was appointed by the Board of Trade to revise the load-lines, Mr. Dunn, although he had retired from the Government service, was asked again to serve on the Committee. His services at the Admiralty, notably when acting as Director of Naval Construction in the absence, through ill-health, of Sir William White, led to the passing of a general minute on the occasion of his retirement, and in this public acknowledgment was made of the "zeal and ability displayed by him in the conduct of his important duties." In 1897 he retired from the Admiralty to join the Board of Messrs. Vickers, Sons and Maxim, who had purchased works at Barrow-in-Furness, and had begun not only the construction of guns and armour, but of ships, propelling machinery, and ordnance machinery of all types. He organized the department and staff of designers, and although he was never a resident in Barrow, he was closely identified with the works. He retired from the directorate a few months before his death, but still continued his interest in naval architecture. On the occasion of the Summer Meeting of this Institution at Barrow in 1901 he read a Paper* on "The Arrangement and Equipment of Shipbuilding Works," in which was given much of the results of his observation and experience. He held a high position in the Engineering Employers' Federation, and his tact and discrimination enabled him to render valuable services in settling differences between the employers and the workmen. He was a Freeman of the Worshipful Company of Shipwrights, and a Vice-President of the Institution of Naval Architects. His death took place suddenly at York, on 17th July 1912, in his seventy-fifth year. He became a Member of this Institution in 1901.

Edwin Frampton was born at Wimborne on 24th August 1849. Having been educated at Wimborne Grammar School, he served

* Proceedings, I. Mech. E., 1901, Part 3, page 555.

an apprenticeship of four years in the Perry Street Works of Mr. J. B. Payne, engineer and millwright, Chard, Somerset, and then went for two years as an improver at the works of Messrs. Ravenhill, Hodgson and Co., Glasshouse Fields, London. During the next three years he acted as engineer for the Imperial Brazilian Collieries at São Jeronymo, Rio Grande do Sul, Brazil, and on his return to England in 1878 he became an active partner in the General Engine and Boiler Co., Hatcham Iron Works, London. This firm became a limited company in 1905, of which he was managing director, a position he held until his death, which took place at Boscombe, from heart failure, on 12th July 1912, in his sixty-third year. He became a Member of this Institution in 1884.

WILLIAM GUTCHER was born at Aberdeen on 26th November 1859, and was educated at the St. Clement's Schools in that city. His apprenticeship was served from 1874 to 1879 at the works of Messrs. Blaikie Brothers, engineers, of Aberdeen, and during the same period he attended technical classes in the evenings at the Mechanics' Institute. On the completion of his apprenticeship he worked for two years as a journeyman engineer, and then in 1882 he was appointed superintendent engineer of the Singapore Oil Mills. In 1900 he was promoted to be works manager, which position he held until his death. This took place very suddenly while at work, on 6th May 1912, in his fifty-third year. He became an Associate Member of this Institution in 1904; he was a Member of the Association of Engineers of Singapore, of which he had been President for five years.

CHARLES BENJAMIN HADENGUE was born in 1863, and was educated at Westbourne College, Notting Hill, London. As his parents were in India, he went out there in 1877, and was apprenticed in the locomotive workshops of the East Indian Railway at Jamalpur, Bengal. On the completion of his apprenticeship in 1882 he was engaged in the same works as assistant draughtsman until 1884, when he became engineer in charge of the Kooldiah

Collieries at Giridi, belonging to the Bengal Coal Co. Three years later he became engineering representative in India of Messrs. Robey and Co., Lincoln, and from 1889 to 1892 he was in charge of the engineering department of the firm of Messrs. Williamson, Magor and Co., Calcutta. In the latter year he was appointed engineer to Messrs. Carew and Co., sugar refiners, Rosa, U.P., which position he held until his death. During this period he carried out many improvements in the buildings and machinery, and also had the inspection duties of the firm's other factories at Katni and Asensole. His death took place very suddenly on 18th June 1912, at the age of forty-nine, of heart failure, in Tasmania, whither he had gone to see an orchard estate which he had purchased in anticipation of retirement in a few years' time. He became an Associate Member of this Institution in 1894.

John William Howard was born at South Hampstead, London, on 25th June 1856, and was educated at University College School and King's College, London. In 1874 he began an apprenticeship at the Great Eastern Locomotive Works, Stratford, under the late Mr. William Adams, and passed through the various shops and drawing office. He also inspected the construction of new locomotives at the works of Messrs. Kitson and Co., and other builders. In 1881 he joined the firm of Messrs. John Spencer and Son, Ltd., of Newburn Steel Works, Newcastle-on-Tyne, as their representative in London. He was afterwards with Mr. Josiah McGregor, whom he assisted in the designing and construction of light draught steamers for India, South America, and other countries. In 1894 he became assistant general manager of the Gloucester Carriage and Wagon Co., and on the retirement of Mr. Alfred Slater in 1904 he was appointed general manager of that company, which position he occupied until July 1911, when he resigned owing to ill-health. During the time he was in Gloucester very important alterations and additions were made to the Wagon Co.'s extensive works, making them one of the finest in the country. In October 1911 he sailed with Mrs. Howard for New Zealand, intending to take a prolonged trip round

the world, but his death took place at Auckland on 14th August 1912 at the age of fifty-six. He became a Member of this Institution in 1882.

HENRY LEA was born in Birmingham on 6th January 1839, and was educated at King Edward's School in that city. Displaying a remarkable gift for mechanics, even while at school, he was apprenticed to Mr. J. E. Hodgkin, who owned a small factory in Berkley Street, Birmingham, which, later on, was acquired by Messrs. May and Mountain. There he obtained a thorough knowledge of the practical side of engineering, and at home he studied the theory. On leaving Messrs. May and Mountain he entered the works of Mr. Walter Williams, at Albion, where he acquired a knowledge of rolling-mill practice and bridge construction. In 1862, at the age of twenty-three, he started business on his own account as a consulting engineer in Birmingham, and gradually built up a large connection. For a time his work covered mostly the various branches of mechanical engineering practice, but in 1882 it was supplemented by electrical work. In that year he carried out the electrical lighting of the Birmingham Town Hall, and he was also associated with Colonel Crompton and Mr. (now Lord Justice) Fletcher Moulton in drafting the first Electric Lighting Order, namely, that for Chelsea. For many years he was honorary consulting engineer to the Birmingham General Hospital, and when the new hospital was erected he designed and put down the electric-motor installation for working the fans in connection with the ventilation scheme. He was responsible for a large number of other plants, among which may be mentioned the high-pressure hydraulic installation for the Birmingham Corporation in Dalton Street; the electric lighting of the Staffordshire County Asylums at Cheddleton, near Leek, and at Burntwood, Lichfield; the Birmingham Corporation Asylum at Hollymoor; the electric plants at the Birmingham Technical School, and at the Meat Market; the electric lighting of Birmingham University, etc. Apart from his business, Mr. Lea's hobby was the making of models of all kinds of machinery at

home, and at the time of his death he was engaged on a model of the latest type of Midland locomotive. He was elected a Member of this Institution in 1860, and served on the Council during 1898–99, and from 1902 until his death. He was one of the Institution's representatives on the Engineering Standards Committee, and was a member of the Sectional Committee on Screw Threads and Limit Gauges. He was also a Member of the Institution of Civil Engineers, and of the Institution of Electrical Engineers. He helped very considerably in the organization of the successful Meeting of this Institution in Birmingham, July 1910, which was held jointly with the American Society of Mechanical Engineers, and acted as one of the Vice-Chairmen of the Reception Committee. He was a member of some of the principal philanthropic institutions in Birmingham, and rendered good service on the committees of the General Hospital and Institution for the Blind. His death took place at his residence in Edgbaston, Birmingham, on 20th July 1912, in his seventy-fourth year.

John Machray Ledingham was born at Aberdeen in 1849. He served his time at the Royal Laboratory, Woolwich, from 1865 to 1870, and then worked as a turner at the works of Maudslay, Sons and Field during 1870 and 1871. In the latter year he was engaged by the Indian Government, and until 1876 he was foreman of fitters and turners at the Small Arms Ammunition Factory, Kirkee, Bombay. In 1876 he returned to England and was appointed foreman at the Royal Laboratory; and in 1885 he became principal foreman. Three years later he was promoted to be assistant manager, and manager in 1890. This position he held until his death, which took place at Woolwich on 22nd July 1912, at the age of sixty-three. He became a Member of this Institution in 1890.

Thomas Templeton Mackie Lumsden was born in Edinburgh on 4th March 1850. Having been educated at Dr. Andrew Thomson's School in that city, he entered at an early age the drawing office

of Messrs. G. and W. Bertram, St. Katherine's Works, Sciennes, Edinburgh. Being a skilful worker, he soon won recognition and was entrusted with a great deal of responsible work in the principal paper mills in the country. In 1884 he left the firm to join with the late Mr. James McFarlane in forming an Engineering Department of the business of Messrs. James Milne and Son, then at Milton House, Canongate, Edinburgh. In the following year the firm moved to extensive new works at Abbeyhill, and Mr. Lumsden became managing director. This position he held until his death, which took place after a long illness, on 29th July 1912, at the age of sixty-two. He became a Member of this Institution in 1895.

WILLIAM BAYLEY MARSHALL was born at Norwich on 17th November 1850, being the eldest son of the late William Prime Marshall,* Secretary of this Institution from 1849 to 1878. After having been educated privately, he served his apprenticeship with Messrs. Dübs and Co., of Glasgow, locomotive builders, and then went to the works of Messrs. Robert Napier and Sons, from 1873–75. For a short time he was in the drawing office of Messrs. Simpson and Co., of Pimlico, and in 1876 he was appointed works manager to the Bridgewater Engineering Co. Two years later he became general manager to the Staffordshire Wheel and Axle Co., and then in 1882 he joined his father, under the title of William P. Marshall and Son, as consulting and inspecting engineers, specializing in inspection of railway rolling stock for the Crown Agents for the Colonies. He acted as Joint Honorary Secretary for the Jubilee Meeting which was held in Birmingham in 1897, and contributed in no small degree to its success. In 1906 he retired from active business, through failing health, and went to reside at Malvern, where his death took place on 23rd July 1912, in his sixty-second year. He became a Member of this Institution in 1877.

JOHN HARLEY MEIKLEJOHN was born at Lasswade, Midlothian, on 4th April 1878. He was educated at Victoria College, Jersey,

* Proceedings, I. Mech. E., 1906, Part 2, page 335.

Channel Islands, and Merchiston Castle School, Edinburgh. After this he took an engineering course at Heriot Watt College, Edinburgh, from 1894 to 1896, and then he was apprenticed for three years to the Fairfield Shipbuilding and Engineering Co., at Govan. On the outbreak of the Boer War he volunteered for service, and worked for a year in the locomotive department, having charge of armoured trains running between Johannesburg and Klerksdorp. In 1902 he went out to the Malay States, where he became assistant engineer in the Suder Sereuban Tin Mine. From 1903 to 1908 he acted as engineer-in-charge at the Rahman Tin Mine, Upper Siam, and on his return to England in 1909 he joined the firm of A. E. Kitsell and Co., engineers and brass founders, Harlesden, London, as managing partner. At the end of 1911 he again went to the Malay States on a short visit to the Rahman Mine, and on his return journey his death took place at sea, on 7th June 1912, at the age of thirty-four. He became an Associate Member of this Institution in 1910.

George Blake Oughterson was born at Liverpool on 22nd November 1837. At the age of seventeen he became an apprentice at the locomotive and carriage works of the Paris and Rouen Railway, under Mr. William Buddicom, and the last year of his apprenticeship was spent in the prime cost and estimating department of these works, an experience which proved of the greatest value subsequently. On the completion of his term he was appointed, in 1858, resident engineer under Sir Donald Campbell on the Lowgill–Ingleton section of the Lancaster and Carlisle Railway. After three months' service there he was, though only just over twenty-one years of age, appointed to the post of assistant locomotive superintendent on the Great Luxemburg Railway, under the late Mr. Thomas Kitson and subsequently the late Mr. Price Pritchard Bailey. This position he held for four and a half years, and then he joined the staff of the late Mr. Edward Preston, who was one of the concessionaires for the Belgian Railway from Tamines to Landen, taking charge of the Brussels office. Here he supervised the plotting off of the sections,

the calculations of the quantities of earthworks, etc., the design of the bridges, etc. In this position he remained two years, when, in 1865, he was offered, and accepted, a partnership in the old-established firm of William Martin, Son and Co., at Rouen. Foundry work, however, proved unattractive, and accordingly William Martin decided to take over adjoining engineering works, and started the manufacture of all stationary railway plant, and also of sugar machinery for export to the French colonies. As success was being attained, the Franco-German War broke out, and the firm had to go into voluntary liquidation at the end of the war, owing to the difficulty of obtaining money. The engineering business was taken over by Messrs. Manlove, Alliott and Co., of Nottingham, on condition that Mr. Oughterson should remain as their manager at Rouen. This arrangement lasted up to the end of 1877, when he became general manager at Mr. Peter Brotherhood's works in London. In this connection he was delegated to superintend and carry out trials in connection with torpedoes and their accessories, and visited on several occasions all the arsenals of the French Government, the Italian arsenal at Spezia, the Dutch arsenal at Halder, and the Danish arsenal at Copenhagen. The association with Mr. Brotherhood ceased in March 1897, and immediately afterwards he joined the late Mr. W. Harry Stanger as manager of the engineering department of his business, as consulting and inspecting engineer to the Crown Agents for the Colonies, to two Admiralty departments, and to several self-governing colonies. His death took place at Folkestone from an attack of cerebral hæmorrhage, on 2nd August 1912, in his seventy-fifth year. He became a Member of this Institution in 1867; and he was also a Member of the Société des Ingénieurs Civils de France.

William Powrie was born at Dundee on 10th October 1840, and was educated at a local school. He served his time from 1854 to 1860 as a millwright, and worked as a journeyman for a few years, during which time he was engaged in the erection of flour mills in different parts of Scotland. In 1863 he went to

Edinburgh and was employed as millwright and pattern maker in the engineering department of the Scottish Vulcanite Co., and this was followed by a period of three years in the works of Messrs. Nasmyth, Wilson and Co., Patricroft, Manchester, which he undertook in order to extend his engineering knowledge. In 1868 he became chief draughtsman and assistant manager to Messrs. Furnival and Co., printers' engineers, Manchester, and remained there for five years, during which period he designed several new kinds of printing machines. In 1873 Messrs. Adam and Co., printers and publishers, of Newcastle-on-Tyne, were building new works for the manufacture of printing machinery, and invited Mr. Powrie to take charge of their engineering department. Their offer was accepted and he remained with them for three years, until the firm went into liquidation. In February 1877 he went to London to open a branch for Messrs. Furnival and Co., and remained as their general manager in London and district up to the time of his death. He always took a great interest in technical education, and frequently lectured on various subjects to classes of students. In 1899 he read a Paper* before this Institution on "Machinery for Book and General Printing." His death took place at his residence in Clapham, London, on 19th July 1912, in his seventy-second year. He became a Member of this Institution in 1898.

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* Proceedings, I. Mech. E., 1899, Part 1, page 103.

# The Institution of Mechanical Engineers.

## PROCEEDINGS.

### October 1912.

The first Ordinary General Meeting of the Session was held at the Institution on Friday, 25th October 1912, at Eight o'clock p.m.; Edward B. Ellington, Esq., President, in the Chair.

The Minutes of the previous Meeting were read and confirmed.

The President announced that the Ballot Lists for the election of New Members had been opened by a Committee appointed by the Council, and that the following one hundred and sixteen Candidates were found to be duly elected :—

MEMBERS.

| | |
|---|---|
| Dawson, Professor Tom Stafford, | Bombay. |
| Elliott, Clifford Francis John, | Smyrna. |
| Foreman, James Thomas Weatherall, | London. |
| Pierce, Robert Cecil, | Cambridge. |
| Price, Ernest, | Manchester. |
| Robertson, Thomas Robert, | Liverpool. |
| Smith, William Gregory, | Ichapur. |
| Turner, Frank, | Woolwich. |

ASSOCIATE MEMBERS.

| | |
|---|---|
| Abbott, John Haelen Abbott, . . | London. |
| Adey, Douglas Francis, . . . | London. |
| Anderson, James Thomas, . . . | Bristol. |
| Armitage, Charles Varley, . . | London. |
| Arney, Arthur Edward, . . . | Perth, W.A. |
| Avent, Ben Stafford, . . . . | London. |
| Barlow, Charles Robert, . . . | Cape Town. |
| Baskerville, Robert Henry, . . | Manchester. |
| Beetham, Hugh Stanley, . . . | Barrow-in-Furness. |
| Bettison, Frank Casson, . . . | Leeds. |
| Beynon, Herbert, . . . . | London. |
| Billinghurst, Raymond William Bowen, | London. |
| Bird, Philip Augustus, . . . | Calcutta. |
| Blythe, Walter Afford, . . . | Sheffield. |
| Boak, Charles Frederick, . . | Calcutta. |
| Brabner, Frederic, . . . . | Leeds. |
| Bradshaw, Stanley, . . . . | Birmingham. |
| Brameld, Phillip, . . . . | Punta Arenas. |
| Bull, Charles Gaston, . . . | Paris. |
| Coates, John, . . . . . | Newcastle-on-Tyne. |
| Codling, John Henry, . . . . | Shipley. |
| Cookson, Frederick Randolph Cecil, . | Hull. |
| Copping, Gilbert Lloyd, . . . | London. |
| Craig, Robert Gordon, . . . | Calcutta. |
| Cuthbert, Fred, . . . . . | London. |
| Damant, Alfred Charles Claude, . | London. |
| Deane, Richard Hill Asperne, . . | London. |
| Donnelly, John, . . . . . | Pretoria. |
| Edye, John de Greet, . . . . | London. |
| Fairbairn-Crawford, Ivo Frank, . . | Newcastle-on-Tyne. |
| Farbridge, Joseph William, . . | Hiogo Ken, Japan. |
| Finch, Rupert James, . . . . | Leicester. |
| Fox, Leonard Monro, . . . . | London. |
| Fraser, John Hill, . . . . | Trincomalee. |
| Furkert, Frederick William, . . | Wellington, N.Z. |

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| Gaunt, John Waugh, . . . . | Cossipore. |
| Goodacre, Ernest John, . . . | Yokohama. |
| Greenfield, John, . . . . | London. |
| Griffiths, Herbert Roberts, . . | London. |
| Guy, Henry Lewis, . . . . | Manchester. |
| Haan, Peter de, . . . . . | Milan. |
| Hally, George, . . . . . | Wellington, Salop. |
| Hammett, Darcy Hesseltine, . . | Cardiff. |
| Hepton, Ernest Stanley, . . . | Hull. |
| Hickson, Charles Hamilton, . . | Perth, W.A. |
| Hutchison, Hugh, . . . . | Singapore. |
| Hyde, James Henry, . . . . | Teddington. |
| Keegan, George Hamilton, . . . | Middelburg, C.C. |
| Kemp, Ernest, . . . . . | London. |
| Kent, William Mallet, . . . | Cyprus. |
| Kewley, Frederick Ernest, . . . | Market Drayton. |
| Lawford, Arthur Niven, . . . | Manchester. |
| Lee, Jesse James, . . . . | Gainsborough. |
| Longsdon, Henry Serle, . . . | London. |
| McCrie, Bertram, . . . . | Glasgow. |
| MacGuckin, Charles John Grahame, . | Newcastle-on-Tyne. |
| Marsh, Clifford Llewellyn, . . | Merthyr Vale. |
| Marshall, John Alexander, . . | Havana. |
| Minett, Albert Ernest Sculthorpe, . | Katha, Burma. |
| Morton, Duncan Anderson, . . . | Vancouver. |
| Mountfort, Louis Frederic, . . | Birmingham. |
| Ogden, Robert William, . . . | Warrington. |
| Pacey, Stewart Oswald, . . . | Smyrna. |
| Perks, Thomas, . . . . . | Warrington. |
| Portway, Robert Cornell, . . . | Bromley, Kent. |
| Prescott, Kenneth Seymour, . . | Ashton-under-Lyne. |
| Rees, Alfred Colston, . . . | Llanelly. |
| Reynolds, Oliver, . . . . | Bolton. |
| Rigby, Henry, . . . . . | Nova Scotia. |
| Robson, George, . . . . | London. |
| Rome, George Harold, . . . | London. |

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| Ryder, Alfred Harold, . . . | Kuantan, S.S. |
| Salway, Alec Dudley, . . . | Natal, Brazil. |
| Shenstone, William McCalman, . . | Nazira, Assam. |
| Simon, Louis John, . . . . | London. |
| Smith, Alexander John, . . . | Tokyo. |
| Smith, Frederick Sexstone, . . . | Villa Maria, Arg. Rep. |
| Spencer, Philip, . . . . . | Negapatam. |
| Stanfield, Joseph Reginald Montague, . | Cardiff. |
| Sugg, Harry Guy, . . . . | London. |
| Sutcliffe, Arnold Riley, . . . | Halifax. |
| Tait, Reginald Arnan, . . . | London. |
| Thom, John, . . . . . | Enfield. |
| Tripp, William Howard Sandberg, . | Walker-on-Tyne. |
| Turner, George Bankart, . . . | Newcastle-on-Tyne. |
| Vine, Horace, . . . . . | Calcutta. |
| Whyte, Andrew Liddell, . . . | Douglas. |
| Wright, James William Ewart Gladstone, | Colchester. |
| Young, Geoffrey, . . . . | São Paulo. |

ASSOCIATE.

| | |
|---|---|
| Clark-Neill, James, . . . . | Glasgow. |

GRADUATES.

| | |
|---|---|
| Calthrop, Keith de Suffield, . . | London. |
| Crooke, Sidney Egerton, . . . | London. |
| Gooch, Stanley John, . . . . | East Molesey. |
| Greenfield, Eric Berand, . . . | London. |
| Hayward, Russell Dallas, . . . | Gainsborough. |
| Hill, Sydney Burgan, . . . | Birkenhead. |
| Jackson, Francis Munton, . . . | Brighton. |
| Lamport, Edward John, . . . | Eastleigh. |
| Latham, Percy Barkley, . . . | Crewe. |
| Liley, Henry Deakin, . . . . | Lincoln. |
| McGill, Ernest William, . . . | Brighton. |
| Nicholl, Harry Wynne, . . . | London. |
| Peppercorn, Arthur Henry, . . | Colwick, Notts. |
| Splitter, Bernard, . . . . | London. |

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| Steele, Robert McAra, . . . | Mossend, Lanarkshire. |
| Unger-Vetlesen, Fredrik Wilhelm George, . . . . . | Christiania. |
| Whipp, Frederick George, . . . | London. |
| Whitaker, John, . . . . | Accrington. |

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The President announced that the following twelve Transferences had been made by the Council:—

*Associate Members to Members.*

| | |
|---|---|
| Christiansen, Albert George, . . . . | Bombay. |
| Davison, Edward James, . . . . . | London. |
| Drinnan, John, . . . . . . | Johannesburg. |
| Fyffe, Andrew Morton, . . . . . | Nelson. |
| Kettle, Laurence Joseph, . . . . | Dublin. |
| Read, George Henry, . . . . . | London. |
| Rodd, William James Paulo, Captain A.O D., . | Haulbowline. |
| Rosevere, Gerald Rhodes, . . . . | Bolton. |
| Schofield, Samuel Dean, . . . . | Shipley. |
| Smith, Professor Cades Alfred Middleton, . | Hong Kong. |
| Smith, Thomas Harold, . . . . . | Cairo. |
| Wilcox, John George, . . . . . | Calcutta. |

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The following Paper was read and discussed :—

"Characteristic Dynamical Diagrams for the Motion of a Train during the Accelerating and Retarding Periods"; by Professor W. E. Dalby, *Member*, of London.

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The Meeting terminated shortly before Ten o'clock. The attendance was 142 Members and 53 Visitors.

# CHARACTERISTIC DYNAMICAL DIAGRAMS FOR THE MOTION OF A TRAIN DURING THE ACCELERATING AND RETARDING PERIODS.

By Professor W. E. DALBY, *Member*, of London.

The author proposes to deal with the subject of this Paper under the following heads:—

1. Fundamental importance of the acceleration period.
2. Tractive-force curves.
3. The characteristic Dynamical Diagram for a particular case:—
    - (a) Scales.
    - (b) The diagram.
    - (c) The accelerating force $f$.
    - (d) Limiting speed.
    - (e) Time-speed curve.
    - (f) Time-distance curve.
    - (g) Kinetic-energy-distance curve.
    - (h) Speed-distance curve.
    - (i) Checks to be applied.
4. General features of the Dynamical Diagram.
5. Dynamometer-car record of the Riviera Express from Paddington.

6. Reduction of the data from the Dynamometer-car record to the curves of the Dynamical Diagram.
7. Braking.
   (a) A wheel-element.
   (b) A vehicle composed of $n$ wheel-elements $m$ of which are braked.
   (c) An engine composed of $n$ dissimilar wheel-elements.
8. Tension on a draw-bar due to unequal braking of engine and train.
9. Characteristic Dynamical Diagram for a train stopping from a speed of 60 miles per hour.
10. Moments of inertia of typical pairs of Wheels and Axles.

| | |
|---|---|
| Engine driving-wheel 73 inches diameter.<br>Engine trailing-wheel 73 inches diameter.<br>Engine bogie-wheel $45\frac{1}{2}$ inches diameter.<br>A wood-centred carriage wheel $45\frac{1}{2}$ inches diameter. | *See* Table 3 (page 914). |

## 1. Fundamental Importance of the Acceleration Period.

The general development of electric traction for the purpose of operating suburban services is largely due to one important difference between the steam- and the electric-locomotive. In the case of the steam-locomotive the power is limited to that of the boiler which the locomotive carries, and this is strictly limited in size by the construction-gauge. There is no such limitation imposed upon the power of an electric-locomotive, since it is connected with and can draw upon the boiler-power installed in a central station, and can therefore temporarily work at a power greatly exceeding the possibilities of a steam-locomotive. The practical consequence of this difference is that, during the starting period where large power is required for short intervals of time, the electric-locomotive (or the electric multiple-unit train) answers to the demand without difficulty, whilst the steam-locomotive reaches the limit of its power at comparatively small accelerations.

In the case of the London, Brighton and South Coast Railway electrified service, for example, during the accelerating period the horse-power touches 1,600, a power quite beyond the capacity of any steam-locomotive which could conveniently be employed on a suburban service. Once the journey speed has been attained, the steam-locomotive has sufficient power to meet all the traffic requirements of local, express passenger, and ordinary and express goods services of the present time. The power of the motor to accelerate rapidly has secured its adoption, or at any rate has largely influenced the electrification of steam services where the intervals between the trains are small and the stops frequent.

The study of the characteristics of the motion of a train during the accelerating period has therefore assumed importance, an importance indicated by the fact that the actual choice of a method of traction for services of a suburban character depends upon the suitability of the tractor to work the train during the accelerating period.

The object of this Paper is to explain a method by means of which time-speed, time-distance, speed-distance, and energy-distance curves may be derived from a curve of tractive force expressed as a function of the velocity, to consider a method of reducing the data obtained from a dynamometer-car record in order to obtain information regarding vehicle and engine resistance, to illustrate by means of a dynamical diagram the principles underlying the practice of braking, and incidentally to consider the question of the energy of rotation stored in the wheels of the train.

One advantage of the method about to be explained is that the accuracy of the curves deduced can be easily checked, and that by the use of the integraph and starting with a tractive-force curve, the whole family may be rapidly drawn.

## 2. Tractive-Force Curves.

The tractive force exerted on a train may be maintained at a nearly constant value by means of electric motors from the start up to the journey speed; and were it not for the fact that the train-resistances increase with the speed, the accelerating force

would be constant and the dynamics of the problem would be simple. The method of constructing the characteristic dynamical diagram will be explained in connection with the tractive-force curve of a steam-locomotive, because the curve is more variable in character than that of an electric-motor, and therefore more points of interest are presented.

The tractive force exerted by a steam-locomotive is a more variable function of the velocity during the starting period than is the case with an electric-motor.

From the start up to some ill-defined speed in the region of 50 revolutions per minute, the tractive force exerted by a locomotive can be maintained at the approximately constant magnitude determined by the weight on the coupled wheels. The tractive force corresponding to the weight on the coupled wheels, exerted at 50 revolutions per minute, roughly corresponds to a rate of working equal to the maximum power of the boiler. The power of the boiler varies somewhat with the speed, yet without serious error the power may be regarded as approximately constant above 50 revolutions per minute, so that as the speed increases the cut-off must be reduced in order that the boiler pressure may be maintained.

As the speed increases, the steam finds increasing difficulty in getting into and out of the cylinders through the pipes, ports, passages, and round the bends, the effect of which is to diminish the i.h.p. which can be exerted at high speeds, since, for a given cut-off and a fixed position of the regulator, the weight of steam which finds its way in the cylinder falls off almost according to a straight-line law as the speed increases.

The curve of maximum tractive force for a steam-locomotive is determined therefore, first, by the weight on the wheels; secondly, by the maximum boiler-power; and, thirdly, at high speeds by the design of the ports, steam-passages, and cylinders.

The first part of the curve is a straight line corresponding to the maximum value of the tractive force calculated from the well-known formula—

$$\mathrm{T} = \frac{0{\cdot}8\, npld^2}{2\mathrm{D}},$$

where $p$ is the boiler-pressure by gauge in pounds per square inch, $n$ is the number of cylinders, $l$ is the stroke, $d$ is the diameter of the cylinders, and D is the diameter of the driving-wheels, $l$, $d$, and D being in inches. Or alternatively from the expression

$$T = W/5,$$

where W is the total weight on the coupled wheels.

The second part of the curve during which the boiler works at its maximum rate is roughly a rectangular hyperbola, and this changes to a straight line sloping towards the speed axis when the piston-speed is about 1,000 feet per minute, which forms the third part of the curve. The point at which the curve changes in character is determined by the design of the ports and steam-passages. With large ports and short straight passages, the change takes place at a higher piston-speed than that mentioned above.

### 3. The characteristic Dynamical Diagram for a particular case.

The simplest way to explain the method is to work out an example in detail. Let the problem be to draw the time-velocity curve; the time-distance curve; the velocity-distance curve; the force-distance curve, and the energy-distance curve from the following data:—

Weight of engine and tender, 115 tons $= W_e$.

Weight of vehicles, 290 tons $= W_v$.

The start to be made from rest along a gradient of 1 in 1,000 up.

Maximum tractive force exerted by the engine, 12 tons.

Maximum indicated horse-power which can be maintained during the accelerating period, 1,200.

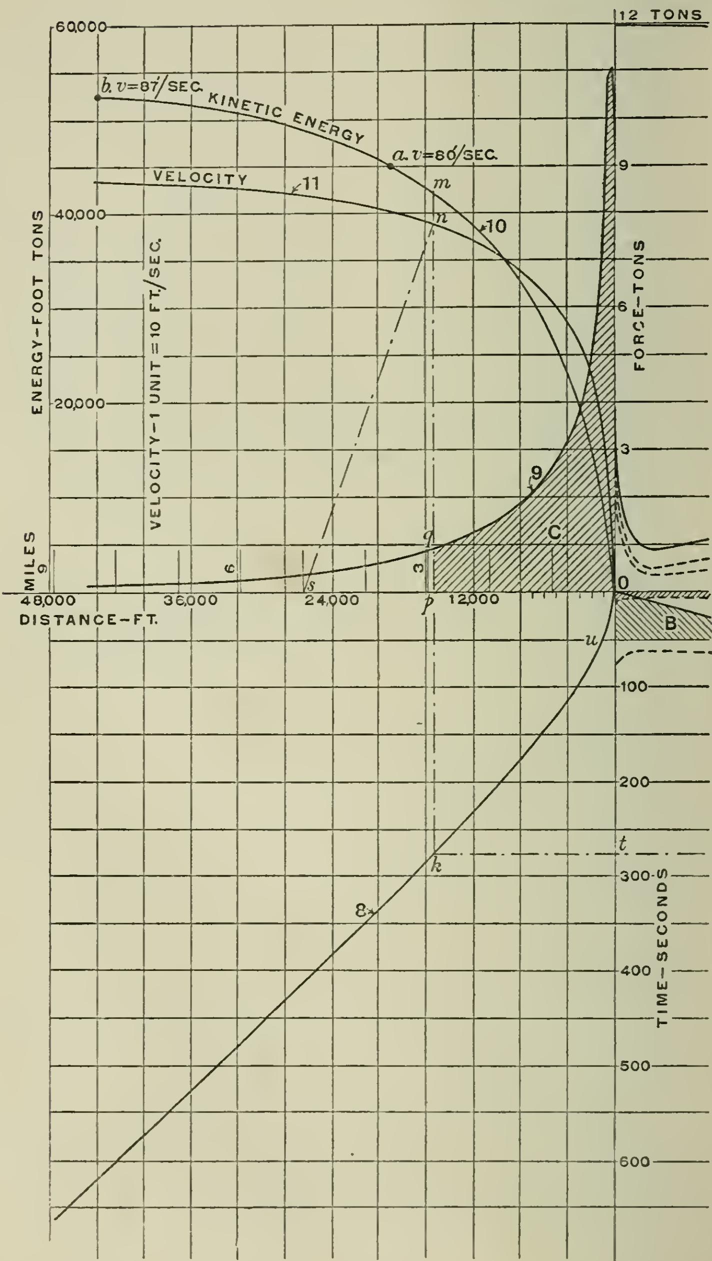
Influence of ports and passages, negligible.

The resistance of the vehicles to be calculated from Mr. Aspinall's formula for a 15-coach train, namely—

$$\text{Total resistance} = W_v\left(2{\cdot}5 + \frac{V^{\frac{5}{3}}}{73}\right) \text{pounds.}$$

The total resistance of the engine and tender to be taken equal to 50 per cent. of the total vehicle resistance, that is to say it is 33 per cent. of the total resistance of the whole train.

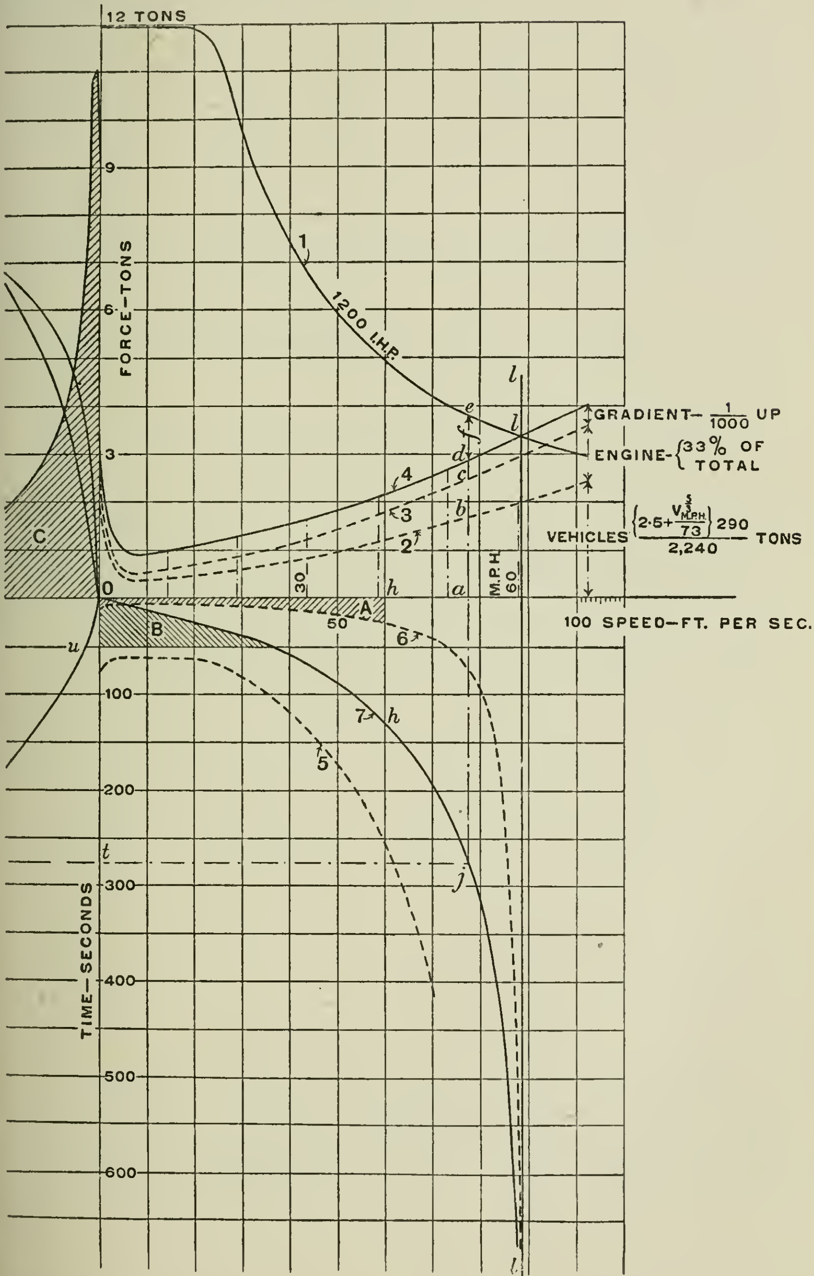
Fig. 1.—*Characteristic Dynamical Diagram for*
Weight of Engine, 115 tons. Weight




(Continued on next page.)

: Train. *Starting from rest on an incline of* 1 *in* 1,000 *up.*
of Vehicles, 290 tons. Weight of Train, 405 tons.

(Continued from opposite page.)



(a) *Scales.*—All the quantities have to be represented in the diagram to scales which are related to one another in a definite way. The most convenient way to deal with the scale relations is to choose some unit of length on the paper and express every quantity in terms of it thus:—

1 unit of length on the paper = $n$ units of the quantity to be represented.

In what follows, for the sake of generality the unit of length on the paper will be referred to as the "unit of length."

(b) *The Diagram.*—Draw two axes at right angles. The part of the horizontal axis to the right of the origin is the *velocity* axis and the part to the left the *distance* axis.

The part of the vertical axis above the origin is the *force* axis and the part below the *time* axis.

The first step is to construct curves of resistance and tractive force, in order to obtain the value of the force which is available for accelerating the train. At a particular speed the force available for acceleration is the difference between the tractive force corresponding to the horse-power developed in the cylinders and the whole tractive resistance including engine friction.

(c) *The Accelerating Force f.*—Plot the two parts of the curve of total tractive force from the data of maximum tractive force and i.h.p. given above. If R represents the total resistance corresponding to the maximum i.h.p. given above when the speed is $v$ feet per second,

$$Rv = \frac{550 \times \text{i.h.p.}}{2{,}240} \text{ ft.-tons per second;}$$

from which

$$R = \frac{550 \times 1{,}200}{v \times 2{,}240} \text{ tons} = \frac{295}{v \text{ ft./sec.}} \text{ tons} \qquad . \qquad . \quad (1)$$

Points on Curve 1, Fig. 1, which is the curve of total tractive force, are plotted from this expression. The curve is continued upwards until the corresponding tractive force is 12 tons, the limit imposed by the load on the driving-wheels. Draw a

horizontal line, therefore, at 12 tons, rounding off the junction between it and the curve plotted from equation (1).

Next plot Curve 2 to represent the tractive resistance of the vehicles. The ordinates are calculated from

$$R_v = 290 \left\{ 2 \cdot 5 + \frac{V^{\frac{5}{3}}}{73} \right\} / 2{,}240 \text{ tons} \quad . \quad . \quad . \quad (2)$$

Plot the total engine-resistance upward from Curve 2 as a base from any formula suitable for the problem. In the present case it is taken equal to half the total vehicle-resistance.

The ordinate $bc$ is, therefore, equal to $\frac{1}{2}\, ab$.

The ordinate $ac$ represents the total train-resistance on the level.

From Curve 3 as base, plot the constant resistance of the gradient. This is $(115 + 290)/1{,}000 = 0 \cdot \dot{4}$ ton approximately.

Curve 4 represents the total train-resistance due to the tractive resistance and the gradient combined. If the gradient is down instead of up, the ordinates corresponding to the gradient resistance must be plotted downwards from Curve 3, so that the curve of total resistance would then fall below Curve 3.

The vertical intercept between Curves 1 and 4 represents the accelerating force $f$ as a function of the speed. Thus the intercept $f$ on the ordinate $ae$ represents the magnitude of the accelerating force when the speed is that represented by $0a$.

(d) *Limiting Speed.*—The point of intersection $l$ of the tractive force Curve 1 and the total resistance Curve 4 fixes the speed at which the accelerating force vanishes. This is the limiting speed towards which the train approaches. The train never actually reaches this limiting speed, but the time taken to reach any assigned speed within the limit as well as the distance travelled by the train during that time can easily be found, as will be seen below.

Draw a vertical $ll$ through the point of intersection $l$. This is an asymptote which the time-speed curve approaches but never reaches. In the diagram the limiting speed for the conditions assumed is just over 60 miles per hour.

(e) *The Time-Speed Curve* 7. — The fundamental dynamical relation between the force $f$, the mass M upon which it acts, and $a$ the acceleration produced is

$$f = \mathrm{M}a = \mathrm{M}\frac{dv}{dt} \quad . \quad . \quad . \quad . \quad (1)$$

Separating the variables

$$dt = \frac{\mathrm{M}}{f}dv \quad . \quad . \quad . \quad . \quad . \quad (2)$$

so that

$$t = \int_0^{v_1} \frac{\mathrm{M}}{v}\, dv \quad . \quad . \quad . \quad . \quad (3)$$

This equation could be integrated directly if $\mathrm{M}/f$ (which is the reciprocal of the acceleration) could be expressed as a continuous function of $v$ reducible to one of the standard forms.

Whatever be the form of the function, it can always be integrated graphically by the following process. First plot $\mathrm{M}/f$. To do this draw a series of ordinates; scale off each value of $f$, and plot the quotient $\mathrm{M}/f$ (quickly found on a slide-rule) vertically downwards from the speed-axis along the corresponding ordinate. Curve 6 is obtained in this way. It is plotted to the scale: 1 unit of length = 10 units $(\mathrm{M}/f)$. The part of Curve 5 from 0 to 60 feet per second is plotted to the larger scale: 1 unit of length = 1 unit of $(\mathrm{M}/f)$. The mass M to be used in these calculations is that equivalent to the actual weight of the train increased by 12* per cent. to allow for the acceleration of the revolving masses; this point is discussed in detail below.

The total weight of the train is 405 tons. The mass M to be used in the calculations is therefore

$$(405 + 48)/g = 14.$$

To integrate these curves graphically, imagine an ordinate to start from the origin 0 and to move to the right. At any instant the axis, the curve and the ordinate will enclose an area like the area marked A on the diagram. This area suitably interpreted with regard to the scales represents the value of the integral in equation (3) between the limits $v = 0$ and the value

---

* This is a maximum allowance.

of the velocity at which the ordinate has temporarily stopped. The area therefore represents the time taken by the train to acquire the speed corresponding to the position at which the ordinate is temporarily stopped.

Curve 5 is used as long as it falls within the limits of the diagram, after which the process is applied to Curve 6.

With regard to the scale on which one unit of area represents seconds—

1 unit of length represents 10 units of velocity.

1 unit of length represents 1 unit of $(M/g)$ on Curve 5.

So that 1 unit of area represents 10 seconds.

Similarly on Curve 6, 1 unit of area represents 100 seconds.

Set the times so found downwards along the proper ordinates, and points on the time-speed curve are found. Thus the area A represents 130 seconds, and this is set down along the ordinate $hh$, thus determining the point $h$ on Curve 7.

The time-scale may be chosen arbitrarily. In the diagram it is taken so that 1 unit of length represents 50 seconds.

(f) *The Time-Distance Curve* 8.—The velocity $v$ is equal to the rate of change of displacement $x$. That is

$$v = \frac{dx}{dt} \tag{4}$$

so that

$$dx = v\,dt \tag{5}$$

from which

$$x = \int_0^{t_1} v\,dt \tag{6}$$

The velocity $v$ as a function of the time has just been found and is represented by Curve 7. Integrate this curve graphically in the way just explained, noting that the area to be integrated lies between the curve and the axis of time. The shaded area marked B represents the value of the integral between the limits $t = 0$ and $t = 50$, and it therefore represents the distance travelled in 50 seconds. This area is set out along the horizontal through the point corresponding to $t = 50$, thus obtaining the point $u$ on the time-distance curve,

Regarding scales,

1 unit of length = 10 feet per second.

1 unit of length = 50 seconds.

Therefore, 1 unit of area = 500 feet. This is plotted to the arbitrary scale of distance: 1 unit of length = 4,000 feet.

By the aid of these two curves, the time required for the train to acquire the speed $v$ and the distance travelled during the time can be read off. For example, the train acquires a speed of 77 feet per second (52 miles per hour) in about 275 seconds and travels over the distance 15,200 feet (2·9 miles) during the period.

(g) *The Kinetic-Energy-Distance Curve* 10.—The work done by the accelerating force, namely,

$$W = \int_0^{x_1} f\,dx \quad . \quad . \quad . \quad . \quad (7)$$

is equal to the energy stored in the train during its motion through the distance $x_1$ feet.

To find points on this curve, replot the accelerating force on the distance axis. To find the point on the distance axis corresponding to a particular value of the accelerating force $f$, continue the direction of the force until it cuts the time-speed curve in $j$; through $j$ draw a horizontal to cut the time-distance curve in $k$; draw a vertical through $k$ and set out the magnitude of $f$ from the point $p$ at which it cuts the distance axis. The force-distance Curve 9 is drawn through points like $q$ found in this way. Integrate this curve graphically, thus obtaining Curve 10. The shaded area marked C represents the value of the integral in equation (7) between the limits $x = 0$ and $x = 15{,}200$ feet, and therefore represents the kinetic energy added to the train during the interval. It is represented by the ordinate $pm$.

Regarding scales,

1 unit of length = 1 ton.

1 unit of length = 4,000 feet.

Therefore, 1 unit of area = 4,000 foot-tons. It is plotted to the arbitrary scale: 1 unit of length = 5,000 foot-tons.

(h) *The Speed-Distance Curve* 11.—This curve is obtained by projecting points on the time-speed Curve 7 horizontally to the time-distance Curve 8, thus fixing positions on the distance axis at which corresponding values of the speed are to be set up. Thus $pn$ is the speed corresponding to the distance 15,200 feet.

The circumstances of the train's motion are thus completely known from this group of curves which together form the characteristic dynamical diagram for the motion of the train.

(i) *Checks.*—Checks should always be applied to test the accuracy of the work and the correctness of the chain of scales used in the construction of the curves. There are two checks which may be easily applied.

The first check is to calculate the energy stored in two different ways. It may be calculated from the expression $Mv^2/2$. It is also given by the ordinates of Curve 10. Hence fixing upon a particular value of the distance, measure off the speed and calculate the corresponding quantity of kinetic energy. This should agree with the quantity scaled off Curve 10. The velocity corresponding to point $a$ on Curve 10 is 80 feet per second. Therefore $Mv^2/2 = 44{,}800$ foot-tons. From the curve it scales 44,700. Similarly at point $b$ the velocity is 87 feet per second and $Mv^2/2 = 52{,}900$ foot-tons. From the curve it scales 52,700. The scales involved in plotting the kinetic-energy curve are the force and distance scales. Those used in plotting the speed-distance curve are the $W/g$ scale, the time-scale, and the distance-scale. Hence the agreement shows that no error has been made in the deduction of the scales.

The second check may be supplied through the theorem that the subnormal corresponding to any point on a speed-distance curve represents the acceleration. Thus since

$$\text{the acceleration} \qquad = \frac{dv}{dt} = \frac{dv}{dx}\frac{dx}{dt} = v\frac{dv}{dx};$$

this latter expression giving the subnormal to a curve of velocity on an $x$ base.

In the diagram $sp$ is the subnormal to the velocity-curve at the point $n$, and it therefore represents the acceleration at the point and therefore the force $f$ when M is known. The dimensions of acceleration in terms of the velocity are $v^2/l$.

1 unit of length = 10 feet per second.
1 unit of length = 4,000 feet.

Hence 1 unit of length of the subnormal represents $10^2/4{,}000$ units of acceleration = $1/40$ foot per second per second, and since the mass is 14, $14/40 = 0{\cdot}35$ ton.

The subnormal $sp$ measures $2{\cdot}8$ units, and therefore represents a force of $0{\cdot}98$ ton. Scaling off the corresponding force it is found to be $0{\cdot}97$ ton, an agreement as near as can be expected.

## 4. General features of the Dynamical Diagram.

The diagram brings out clearly how difficult it is to fix the limiting speed of a particular train. There would be no difficulty if the engine resistance and the vehicle resistance were known accurately, and if in addition the indicated horse-power developed in the cylinders of the locomotive could be predicted with accuracy. But these quantities are all difficult to determine, even with approximate accuracy, and each is subject to large accidental variations.

Another point to notice is the slowness with which the speed increases in the neighbourhood of the limiting speed. For instance, in the diagram half the limiting speed—namely, 30 miles per hour—is attained in 65 seconds, but it takes about 330 seconds to reach 55 miles per hour. The diagram brings out the necessity for engines with large powers of acceleration, even for express services, so that the time required to attain the running speed may be reduced to a minimum.

The same point is illustrated by Curve 11, which shows the velocity plotted on a distance-base. The speed rapidly increases whilst the train is passing over the first mile, but then the curve gets flatter and flatter and the increase of speed per mile gets rapidly smaller and smaller. A speed of 55 miles per hour is

acquired whilst passing over a distance of about $3\frac{1}{2}$ miles, but the increase is very slow afterwards. The group of Curves 7, 8, 9, 10, 11 are all related to one another, so that, given any one of them, all the others may be deduced from it. In the case considered above the curves were derived from the accelerating force $f$ exerted in the engine cylinders. If any one of the curves were given, this accelerating force could be derived from it. Curve 7 was derived by a process of graphical integration from Curve 6; and Curve 8 was obtained by graphical integration from Curve 7. Similarly Curve 10 was obtained by graphical integration from Curve 9.

To derive the curves in the reverse order, namely, 9 from 10 or 7 from 8 or the $f$ curve from 7, the process of graphic integration must be replaced by a process of differentiation. As a graphical process, differentiation is not so satisfactory as integration. But by the application of a mixed analytical and graphical process, results of great practical value may be obtained. These processes have a practical importance in connection with the reduction of results obtained with a dynamometer-car.

## 5. Dynamometer-car record of the Riviera Express from Paddington.

By the kindness of Mr. George J. Churchward, Member of Council, the author is able to illustrate the application of the method to the record of the 10.30 a.m. Riviera Express from Paddington, taken on 30 Nov. 1908. On this date the weight of the vehicles amounted to 289 tons when the train left Paddington. The engine "Dog Star," a four-cylinder simple, type 4–6–0, weighed together with the tender 115 tons.

A portion of the dynamometer-record is shown full size in Fig. 2 (pages 892–3). The paper is drawn under the recording pens at the rate of 1 foot of paper per mile run of the train. A wheel on the dynamometer coach is provided to operate the winding mechanism, and this wheel was accurately turned and was calibrated by running the car along a measured mile. It will be seen that there are seven lines on the diagram above the datum line.

(Continued on next page.)

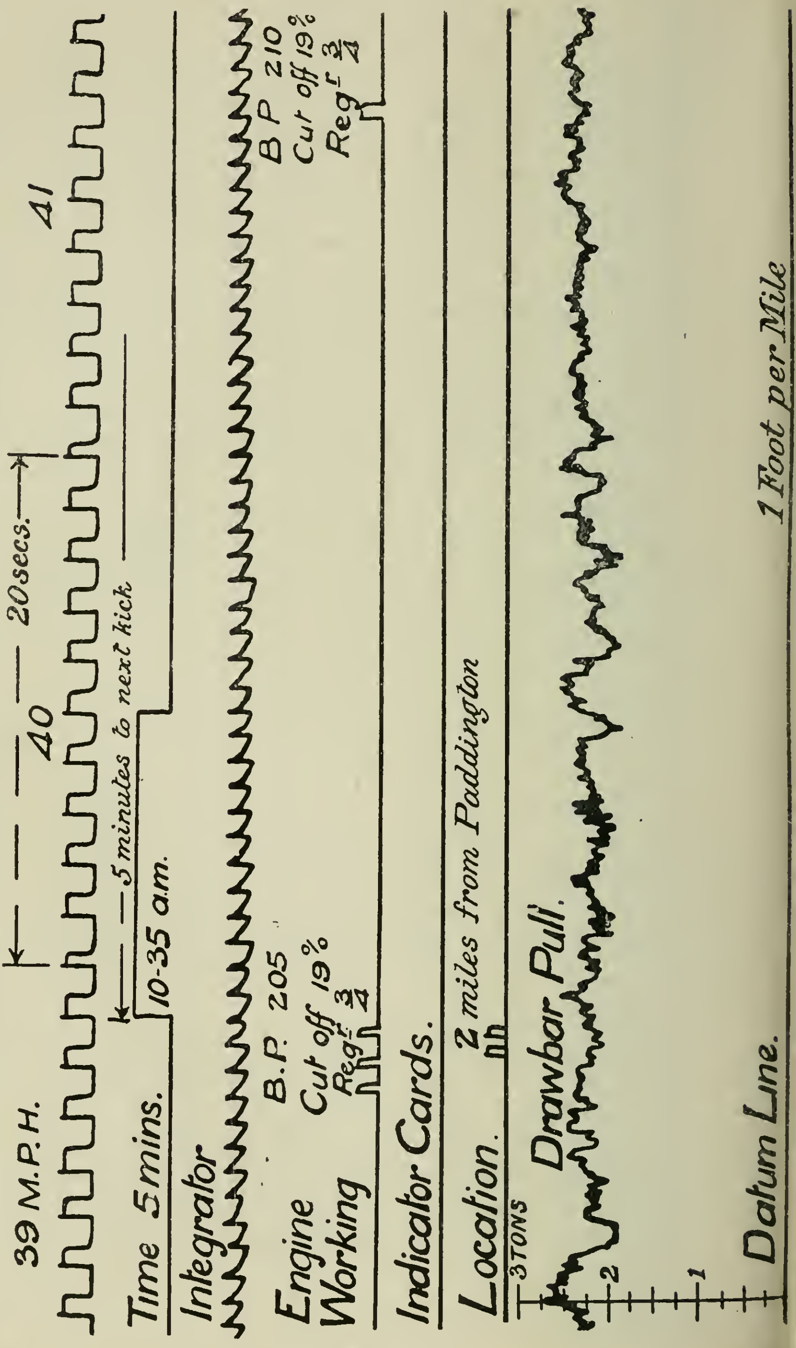



Fig. 2.—*Facsimile of a portion of the Dynamometer-Car Record from which the Time-Distance Curve, Fig. 3, is produced.*

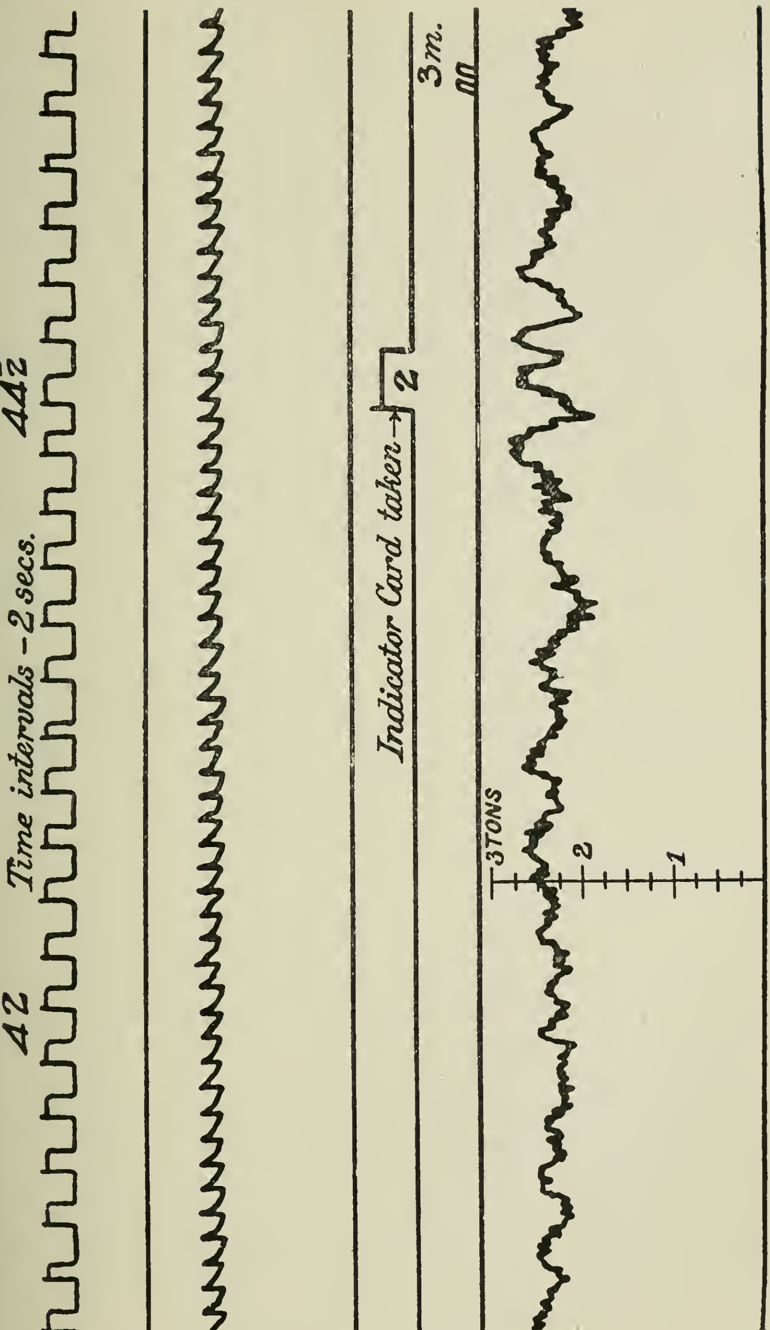



Fig. (continued).—*Facsimile of a portion of the Dynamometer-Car Record from which the Time-Distance Curve, Fig. 3, is produced.*

(Concluded from opposite page.)

Reckoning from the datum line, the first line shows the draw-bar pull.

The second line is traced by a pen normally at rest, but an observer in the car can produce a kick of the pen by pressing an electric button. In this way the location of the mile posts may be recorded, or the position of a station or signal-box.

Kicks of the pen tracing the third line are produced at the time an indicator diagram is taken by a member of the indicator-staff on the engine. The diagram shows the kick corresponding to the taking of No. 2 indicator card.

The fourth line records particulars of the working of the engine. An observer on the footplate notes any change in the conditions of working and at the same time produces a kick of the pen tracing the line, and in order to identify the note in his book with the kick on the record, he signals one kick, two kicks, up to four kicks. Thus it will be seen that, when the note was made that the boiler-pressure was 205 lb. per square inch, and that the regulator was $\frac{3}{4}$ open with a cut-off of 19 per cent., the time of taking the note is indicated on the record by three kicks. The next signal from the engine is indicated by one kick.

The fifth line is an indication that the integrating mechanism is at work. The integrating mechanism is a roller and disk integrator, and shows the amount of work done from the start up to any point of the run.

The sixth line is traced by a pen connected with a clock in such a way that the pen kicks at intervals of 5 minutes.

The seventh line is also traced by a pen connected with the clock, but the pen kicks at intervals of 2 seconds. Since the paper moves under the pen at the rate of 1 foot per mile, the speed of the train is easily deduced from this line.

The record for the complete run of the train is about 300 feet long. The portion shown corresponds to about one foot of the record or one mile of the run.

The complete record indicates a gradual and continuous increase of the speed from Paddington until it reached 70 miles per hour. The train then ran steadily at this speed until required to slow

down for a junction. The diagram, Fig. 3 (pages 896–7), relates to the acceleration period only.

### 6. Reduction of the Data from a Dynamometer-car record to the curves of the Dynamical Diagram.

The time-distance Curve A in Fig. 3 has been plotted from the data of the complete record of which Fig. 2 is a part. This curve can be plotted with accuracy, since corresponding values of the time and distance are deduced easily from the record with negligibly small error. The velocity-time Curve B may be derived from this curve by a process of graphical differentiation. Thus since $v = dx/dt$, and since this ratio also represents the slope of the curve, the velocity corresponding to the point P, say, may be found by drawing the tangent to the curve at P and then computing its slope. At P

$$v = \frac{dx}{dt} = \frac{\text{PN}}{\text{NQ}} = \frac{21{,}120}{294} = 72 \text{ feet per second.}$$

This velocity is set out at NU, and a point U on the velocity-time Curve B is thereby obtained.

This method is not susceptible of much accuracy, since large errors in the magnitude of $dx/dt$ are produced by small errors in the inclination of the tangent, and it is almost impossible to draw a tangent to a point on a curve without error in the inclination, and in the region of the origin, with the type of curve involved in the problems under consideration, the errors become so large that the method is almost valueless.

A better way of deriving B from A is to take a series of horizontals at a distance apart corresponding to some suitable time-interval, and then compute the ratio of corresponding steps of distance $dx$ and time $dt$. The ratio found will be the average value of the velocity during the time-interval corresponding to the distance step $dx$. Thus referring to Fig. 3 the distance step $dx$ during the time-interval $dt$ from 200 to 220 seconds is 920 feet. The average velocity during the interval is then

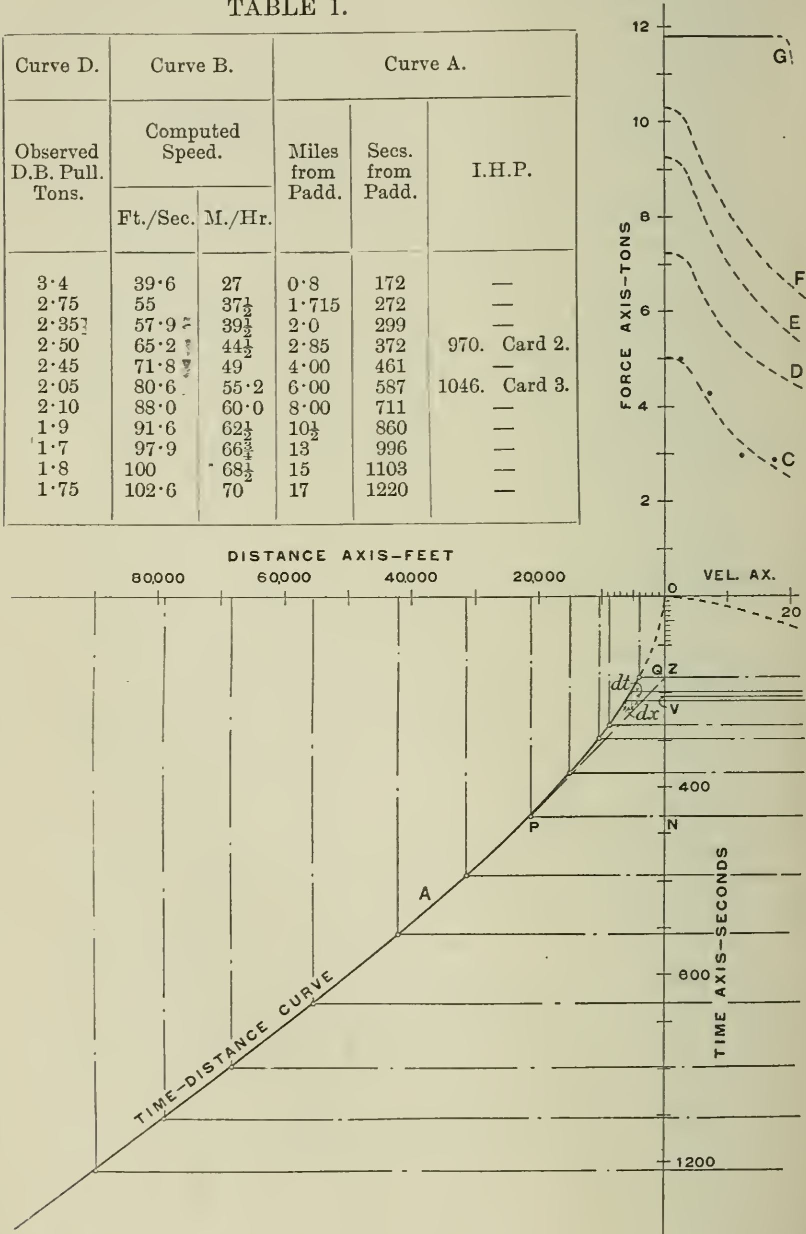
$$\frac{dx}{dt} = \frac{920}{20} = 46 \text{ feet per second.}$$

Fig. 3.—*Characteristic Dynamical Diagram* (*deduced from the Dynamometer-Car*

Engine " Dog Star " 4–6–0. 4 Cylinder Simple. Boiler Pressure, 225 lb. per sq. in.
Each Cylinder, $14\frac{1}{4}$ inches × 26 inches.

TABLE 1.

| Curve D. | Curve B. | | Curve A. | | |
|---|---|---|---|---|---|
| Observed D.B. Pull. Tons. | Computed Speed. Ft./Sec. | Computed Speed. M./Hr. | Miles from Padd. | Secs. from Padd. | I.H.P. |
| 3·4 | 39·6 | 27 | 0·8 | 172 | — |
| 2·75 | 55 | $37\frac{1}{2}$ | 1·715 | 272 | — |
| 2·35 | 57·9 | $39\frac{1}{2}$ | 2·0 | 299 | — |
| 2·50 | 65·2 | $44\frac{1}{2}$ | 2·85 | 372 | 970. Card 2. |
| 2·45 | 71·8 | 49 | 4·00 | 461 | — |
| 2·05 | 80·6 | 55·2 | 6·00 | 587 | 1046. Card 3. |
| 2·10 | 88·0 | 60·0 | 8·00 | 711 | — |
| 1·9 | 91·6 | $62\frac{1}{2}$ | $10\frac{1}{2}$ | 860 | — |
| 1·7 | 97·9 | $66\frac{3}{4}$ | 13 | 996 | — |
| 1·8 | 100 | $68\frac{1}{2}$ | 15 | 1103 | — |
| 1·75 | 102·6 | 70 | 17 | 1220 | — |

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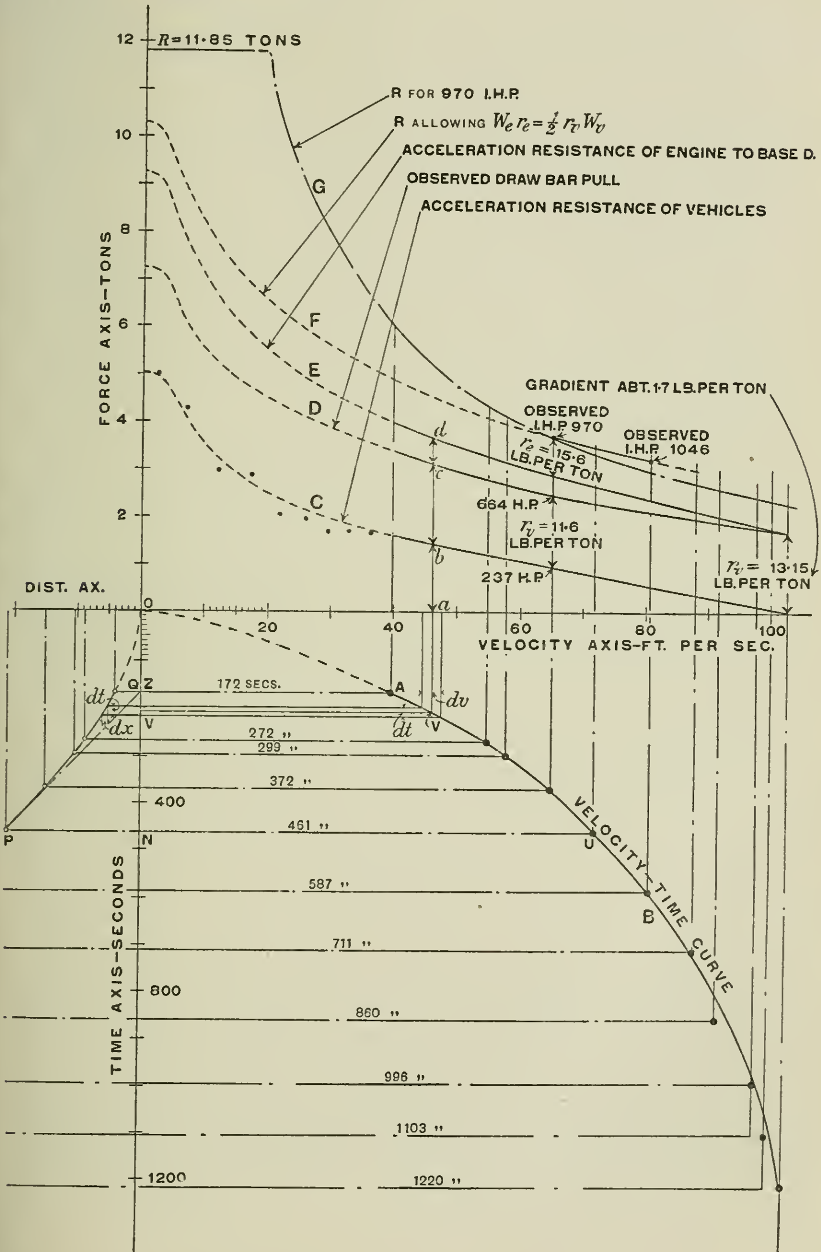
*Record) for the Riviera Express*, 10.30 *a.m. ex Paddington, on* 30 *November* 1908.
Weights, Engine and Tender, 115 tons. Vehicles, 289¾ tons to Westbury.
Coupled Wheels, 6 feet 8½ inches diameter.



This is set out at VV, the horizontal passing through the point corresponding to 210 seconds. V is then a point on the velocity-time curve. The corresponding steps at a particular point of time can however be more accurately computed from the dynamometer-record itself, unless the time-distance curve is set out to a large scale. The dots on the velocity-time curve were fixed by calculation from the data on the record.

Observations were not taken before the speed reached about 40 feet per second. The first part of the time-distance curve and consequently the time-velocity curve are therefore not known from observation. The part of the velocity-time curve between the origin and 40 feet per second can however be drawn in with fair accuracy by the following method.

Two things are known about this part of the curve: first, its direction at A; and, secondly, that at the end of 172 seconds the train has actually travelled 4,224 feet, so that the area OAZ represents the distance, 4,224 feet. The records show that there was no stop, so that the motion and therefore the curve are continuous. Hence, draw a trial curve passing through the origin and passing smoothly into the established part of the curve ending at A. Then measure the area OAZ. If it is found to be 4,224, the first trial has been successful. If there is error, the shape of the curve must be altered to increase or reduce the area.

Since the acceleration is given by $dv/dt$, its value, corresponding to a point of the velocity-time curve B, can be found by either of the methods just explained, preference being given to the second method. By way of example, calculate the acceleration at the point V. During the time interval, from 200 to 220 seconds, the velocity changes 3 feet per second. Hence acceleration $= 3/20$ feet per second per second. This applies equally to the vehicles and the engine. The pull on the draw-bar at this instant producing the acceleration is found by multiplying this by the mass of the vehicles. Allowing for revolving masses the mass is 10 $W/g$ units, so that the corresponding draw-bar pull is $1\cdot5$ ton. This force is represented by *ab* in the diagram, and *b* fixes a point on Curve C which is sketched in to show the draw-bar pull

required to overcome the resistance to acceleration during the whole of the period from the start to the region of uniform speed. The uniform speed attained is, from the records, 70 miles per hour. At this point, therefore, the accelerating force vanishes.

The observed magnitudes of the draw-bar pull at a series of speeds are given in Table 1 (page 896), and Curve D is plotted from these data. The part of the curve which is dotted was fitted in, allowing 17 lb. per ton at the start and then joining it on to the established curve at 40 feet per second.

The vertical distance between the Curves C and D corresponding to a particular speed represents the total resistance of the vehicles at that speed. Thus at a speed of 65 feet per second the total resistance is about 1·5 ton corresponding to 11·6 lb. per ton. At this speed the resistance to acceleration is about 1 ton. The draw-bar pull recorded is 2·5 tons. Using Curve D as a base, plot Curve E to represent the resistance to acceleration of the engine. The mass of the engine, allowing for revolving parts, is approximately $4\ W/g$ units. Thus any ordinate as *cd* is 4/10 of the corresponding ordinate *ab*.

If data were available for plotting the Curve F, which represents the tractive force corresponding to the i.h.p., the vertical distance between Curves E and F at a particular speed would represent the engine-resistance at that speed. Two values of the tractive force equivalent to the i.h.p. are plotted at the speeds at which the diagrams were taken, namely, at 65 feet per second and at 80 feet per second calculated from

$$R = \frac{550 \text{ i.h.p.}}{2240\, v} \text{ tons.}$$

$v$ is here in feet per second.

In the one case where the i.h.p. was 970, R is 3·65 tons, and in the second case where the i.h.p. is 1046, R is 3·2 tons. The engine-resistance at 65 feet per second reduces to 15·6 lb. per ton. The total engine-resistance at this point is roughly about half the total resistance of the vehicles. The Curve F is sketched in on the assumption that this ratio is approximately true at all speeds. It will be understood that the dotted parts of the Curves A, B, C, D,

E, and F are conjectural, but that the dotted part of B is so far correct that the corresponding area does represent the distance travelled by the train in the time 172 seconds, as it ought to do, and that its direction at A is correct. A Curve G of tractive force corresponding to the constant horse-power 970 is drawn on the diagram, and is continued to meet the maximum tractive pull which the engine can exert, namely, 11·85 tons. Comparing Curve G with Curve F, it will be seen that although the dotted part of F is conjectural, yet it is sufficiently correct to show that the engine had a large reserve and, if necessary, could have accelerated the train at a greater rate, or could have equally accelerated a heavier train at the same rate.

The advantage gained by reducing the dynamometer-car records in this way is that accidental errors are smoothed out, and that a comprehensive view is obtained of the working of the engine, and, moreover, the vehicle-resistance can be deduced with greater accuracy from the curves than from isolated calculations. If, also, it were possible to get reliable values of the i.h.p. at a series of speeds, the engine-resistance could be deduced as well.

The resistance of the vehicles and of the engine are seen to be low at the high speeds, and the author considers that the performance of the locomotive is remarkable. To exert a draw-bar pull of $1\frac{3}{4}$ ton at 70 miles per hour (*see* Table 1, page 896), corresponding to 730 horse-power at the draw-bar, is an achievement in locomotive design which it would be difficult to surpass within the limits of the British loading-gauge.

### 7. Braking.

There is a characteristic diagram corresponding to the braking period of the same general type as that for the period of acceleration. The forms of the curves now depend mainly upon the pressure between the brake-blocks and the tyres, that is to say upon the pull exerted by the brake-rods on the brake-levers. Since this pull depends entirely upon the way in which the driver regulates the pressure in the air-pipes of the train, the curve representing the pull as a function of the velocity is about as arbitrary in form as a

curve can well be. Nevertheless there are some useful fundamental principles in connection with the braking of a train which are brought out by drawing a diagram from an assumed form of the curve.

When steam is shut off, the motion of the train is continued against the resistances by the gradual exhaustion of the stock of kinetic energy stored in the train. The principle of the conservation of energy applied to this problem may be stated thus:—

The rate at which the kinetic energy of the train is reduced = the rate at which it is changed into potential energy by a gradient + the rate at which it is transformed into heat by friction.

(a) *Braking of a Wheel-Element.*—At first it is convenient to restrict one's attention to a wheel-element composed of a wheel carrying a load. The total weight of the wheel-element is the weight which would be recorded against the wheel if the vehicle stood on a weigh-bridge provided with a separate steelyard for each wheel. For practical purposes this weight may usually be regarded as equal to the total weight of the vehicle divided by the number of wheels supporting it. Let this weight be W tons. Let I be the moment of inertia of the wheel about its axis; $r$ its radius; $k$ its radius of gyration.

The total store of kinetic energy in the wheel-element is

$$E = \frac{Mv^2}{2} + \frac{I\omega^2}{2} \quad . \quad . \quad . \quad . \quad (1)$$

In this expression the first term represents the energy of translation of the mass $M = W/g$, and the second term represents the rotatory energy stored in the wheel.

The rate at which this stock of energy is reduced is found by differentiating the expression with regard to the time thus:—

$$\frac{dE}{dt} = M\dot{v}v + I\dot{\omega}\omega \quad . \quad . \quad . \quad . \quad (2)$$

This is the rate at which energy is used to drive the wheel-element against the resistances. In an actual train every wheel temporarily becomes a driving-wheel, just as though it formed part of a locomotive, but, instead of being driven by energy derived from the

fuel, it is now driven by energy withdrawn from the limited stock stored in the train by virtue of its motion.

The rate at which energy is transformed into heat by resistances of the nature of friction reduced to the wheel of the element may be analysed into three terms, namely :—

(1) *The rate at which work is done against the natural resistances to the motion of the wheel-element.*—If W is the weight of the element and $r_v$ the resistance to motion in pounds per ton, $\frac{W r_v r}{2{,}240}$ is the couple in foot-tons against which the wheel is turned, and the rate at which work is done is $\frac{W r_v r \omega}{2{,}240} = b\omega$, where $b$ represents the couple and $\omega$ the angular velocity of the wheel.

(2) *The rate at which work is done against the frictional resistance of the brake-blocks.*—If a brake-block is pressed against the tyre with a pressure P in tons, and $\mu_2$ is the coefficient of friction between the block and the tyre, the corresponding frictional couple opposing rotation is $P\mu_2 r$, and the rate at which energy is expended against this is $P\mu_2 r\omega = B\omega$, where B represents the couple due to the application of the brake-block.

(3) *The rate at which work is done against the frictional resistance at the tread as the wheel slips.*—If $v$ is the speed of the train in feet per second, $v/r$ is the corresponding angular velocity of the wheel if it rolls along without slipping, and therefore $(v/r - \omega)$ is the relative velocity of slip against the couple $W\mu_1 r$, where $\mu_1$ is the coefficient of friction between the tyre and the rail; hence the rate at which energy is expended against this is

$$W\mu_1 r\,(v/r - \omega) = s\,(v/r - \omega),$$

where $s$ is the resisting couple which is brought into action when slipping takes place.

*Finally, the rate at which energy is transformed into potential energy* by a gradient rising one foot vertically for G feet horizontally is $Wv/G$ foot-tons per second. The energy equation is therefore

$$Mv\dot{v} + I\omega\dot{\omega} = \omega(B + b) + s(v/r - \omega) \pm Wv/G \quad . \quad (3)$$

In this equation $\omega$ is the angular velocity of the wheel assumed

to be slipping, and $v$ is the speed of the train. If the wheel runs without slipping $v/r = \omega$; $\dot{v}/r = \dot{\omega}$ and the equation reduces to

$$M\dot{v} + I\dot{v}/r^2 = (B/r + b/r) \pm W/G \quad . \quad . \quad (4)$$

In both (3) and (4) the + sign is to be used before the last term if the train is running up a gradient, and the − sign if running down.

The couple B may be regarded as the sum of two couples, namely, one, represented by $B_1$, which produces a frictional resistance against which the linear energy of the train-element is reduced; the other, $B_2$, producing the frictional resistance against which the rotational energy of the wheel is reduced; and further linear motion may be separated from the angular motion, so that

$$M\dot{v} = B_1/r + b/r \pm W/G \quad . \quad . \quad . \quad (5)$$

and

$$I\dot{v}/r = B_2 \quad . \quad . \quad . \quad . \quad (6)$$

Considering equation (5) the terms $B_1/r$ and $b/r$ cannot together be greater than $W\mu_1$, because the motion of the wheel-element being maintained by the exhaustion of the linear energy against these resistances, the wheel will begin to slip immediately the resistance reduced to the rail is greater than $W\mu_1$.

Hence

$$M\dot{v} = W\mu_1 \pm W/G \quad . \quad . \quad . \quad (7)$$

is an equation which gives the maximum retardation $\dot{v}$ which can be applied to the train.

Let $P_1$ denote the pressure applied to the brake-blocks corresponding to the couple $B_1$, and $P_2$ to the couple $B_2$ so that

$$\frac{B_1}{r} = P_1\mu_2$$

and

$$\frac{B_2}{r} = P_2\mu_2$$

and as shown above

$$\frac{b}{r} = \frac{Wr_v}{2{,}240}.$$

Then combining equations (5) and (7) with these relations

$$W(\mu_1 \pm 1/G) = P_1\mu_2 + Wr_v/2{,}240 \pm W/G \quad . \quad (8)$$

From which $P_1$, the pressure which may be applied to the brake-block for the reduction of the linear energy alone, is

$$P_1 = \left\{\mu_1 - \frac{r_v}{2,240}\right\} \frac{W}{\mu_2} \quad . \quad . \quad . \quad (9)$$

Similarly from equation (6)

$$P_2 = I\dot{v}/\mu_2 r^2 \quad . \quad . \quad . \quad (10)$$

The total pressure corresponding to the couple B is then $P = P_1 + P_2$ and this is the value of P when the wheels are just on the point of slipping.

In applying these equations, the maximum retardation consistent with an appropriate value of $\mu_1$ and a given gradient is first to be calculated from (7). This value substituted in (10) together with an appropriate value of $\mu_2$ gives the value of $P_2$. $P_1$ is calculated from (9).

If a pressure is applied to the brake-blocks larger than P the wheel would slip. A smaller value would produce less than the possible maximum retardation, but would provide a margin against slipping.

It will be observed that the instantaneous value of P depends upon four quantities of a variable nature, namely:—

$\mu_1$ the coefficient of friction between the tyre and the rail.

$\mu_2$ the coefficient of friction between the brake-blocks and the tyres.

$r_v$ the train-resistance.

G the gradient.

With regard to $\mu_2$, its value depends not only upon the velocity of rubbing between the block and the tyre but upon the time. Thus, quoting from the classical experiments of Sir Douglas Galton, the observed coefficient of friction between a cast-iron brake-block and a steel tyre when the brake was applied to the wheels of a train kept moving at an approximately uniform speed of 20 miles per hour was 0·18. After 5 seconds this had fallen to 0·15, and after 5 seconds more to 0·13, whilst 20 seconds after application it had

fallen to $0 \cdot 1$. Other values are given in Table 2 (page 912), which is quoted from the Proceedings * of this Institution.

The static value of $\mu_1$, the coefficient of friction between the wheel and the rail, is variable, but for most purposes may be taken at $0 \cdot 2$. If the wheel is skidded, the value of the coefficient of friction is a function of the velocity of sliding, but is not so markedly affected by the time as in the case of brake-blocks. At 60 miles per hour Sir Douglas Galton found that the coefficient of friction of a steel tyre skidded on a steel rail was about $0 \cdot 027$, increasing gradually as the speed decreased, until just before stopping it was $0 \cdot 242$.

The quantity $r_v$ is a function of the speed, and G is variable, but the effect of both quantities is small in comparison with the resistance produced by the brake-blocks.

(b) *A vehicle composed of* n *similar wheel-elements of which* m *are braked.*—A vehicle may be regarded as composed of a number of wheel-elements, and if each one of them is braked the equations above apply to the vehicle as a whole, that is to say the maximum retardation is the value $\dot{v}$ calculated from equation (7), and equations (9) and (10) furnish the corresponding values of $P_1$ and $P_2$, remembering that in these equations W is the total weight of the vehicle divided by the number of wheels and I is the moment of inertia of one wheel. The size of the brake-levers and the brake-cylinder may then be proportioned to the maximum value of P required. In some cases, however, some of the train-elements forming a vehicle are not provided with brake-blocks.

Consider the case of a vehicle composed of $n$ similar wheel-elements, $m$ of which are braked, so that the retardation of the $n$ elements is determined by the brake-power on $m$ of them.

With regard to the $m$ braked elements, equation (7) gives

$$m\mathrm{M}\dot{v} = m(\mathrm{W}\mu_1 \pm \frac{\mathrm{W}}{\mathrm{G}}) \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

With regard to the $(n-m)$ unbraked elements, the only retarding forces are $\frac{\mathrm{W}r_v}{2{,}240}$ the train resistance and the gradient resistance,

* Proc. I. Mech. E. 1878, page 590.

and assuming the train resistance to act at the tread of the wheel, and that both it and the gradient act to reduce the linear energy only

$$(n-m)\mathrm{M}\dot{v} = \left(\frac{\mathrm{W}r_v}{2{,}240} \pm \frac{\mathrm{W}}{\mathrm{G}}\right)(n-m) \quad . \quad . \quad . \quad . \quad . \quad (12)$$

Adding these two equations together and solving for $\dot{v}$, the maximum retardation is given by

$$\dot{v} = \frac{\left\{m\mu_1 + (n-m)\frac{r_v}{2{,}240} \pm \frac{n}{\mathrm{G}}\right\}g}{n} \quad . \quad . \quad . \quad . \quad (13)$$

In most cases this can be simplified to

$$\dot{v} = \frac{m\mu_1 g}{n} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

since the second terms in the numerator are small in comparison with the first term. For example, suppose an 8-wheeled vehicle to consist of eight similar wheel-elements and that the total weight is 32 tons. Further, assume that four wheels only of the eight are braked so that $n = 8$ and $m = 4$. Also let $r_v$ be taken constant at 12 lb. per ton, $\mu_1 = \frac{1}{5}$, and let the stop be made on a rising gradient of 1 in 300, so that $\mathrm{G} = 300$. Then from (13)

$$\dot{v} = \frac{(4/5 + 12 \times 4/2{,}240 + 8/300)32}{8} = \frac{(0\cdot8 + 0\cdot02 + 0\cdot026)32}{8}$$

$$= 3\cdot4 \text{ feet per second per second.}$$

Calculated from (14) $\dot{v} = 3\cdot2$ feet per second per second.

This shows that, in cases where the brake-power applied is near the maximum, the maximum retardation can be calculated from (14) without serious error.

(c) *An engine composed of* n *dissimilar wheel-elements.*—When a vehicle is made up of dissimilar wheel-elements, equation (14) may be put in a more convenient form, a form which in practice applies mainly to locomotives.

Let W be the total weight of the engine and tender.

$w$ be the weight on the unbraked wheels of the engine and tender.

$$\text{Then } \dot{v} = \frac{g(\mathrm{W} - w)\mu_1}{\mathrm{W}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

The instantaneous pressure P to be applied to each wheel can then be found by first calculating $P_1$ from equation (9), substituting for the W therein the weight on the wheel together with appropriate values of $\mu_1$ and $\mu_2$, and then adding to this $P_2$ calculated from (10), using the value of $\dot{v}$ found from (15).

The practical difficulty in connection with these calculations is that $\mu_2$ is a function of the speed, and therefore to maintain the maximum conditions, namely, that $(B + b)$ should be equal to $W\mu_1$, the pressure P must be reduced as the speed decreases. This is illustrated in the characteristic diagram (page 910).

### 8. Tension on a Draw-Bar due to unequal braking of Engine and Train.

When two unequally-braked vehicles are coupled together, the application of the brake produces a tension in the draw-bar (or a compression of the buffers) which in extreme cases may be serious enough to cause the draw-bar to break or to cause a derailment.

Consider two vehicles of mass A and B respectively coupled together. Let $a$ be the retardation which would be produced by the application of the brakes to the vehicle A alone. Let $b$ be the retardation which the application of the brakes would produce on vehicle B alone. Let $c$ be the common retardation actually produced when the vehicles are coupled together, and let T be the tension in the draw-bar.

When the brakes are applied, the force actually acting on the vehicle A to retard its motion is $Aa$ due to the application of the brakes and T due to the draw-bar tension, so that

$$Aa + T = Ac \quad . \quad . \quad . \quad . \quad (1)$$

Similarly, the force acting to retard the motion of B is $Bb$ minus the draw-bar tension, so that

$$Bb - T = Bc \quad . \quad . \quad . \quad . \quad (2)$$

Eliminating $c$

$$T = \frac{AB\,(b - a)}{A + B} \quad . \quad . \quad . \quad . \quad (3)$$

and eliminating T

$$c = \frac{Aa + Bb}{A + B} \quad . \quad . \quad . \quad . \quad (4)$$

In these equations A and B are respectively the weights of the vehicles divided by $g$.

By way of example, calculate the tension on the draw-bar between the engine and the tender corresponding to maximum retardation on the level in the case of an engine of the 4–4–0 type with bogie-wheels unbraked, coupled to a tender and a train with all the wheels braked, from the following data:—

Total weight on bogie-wheels = 16 tons. Total weight of engine = 46 tons. Weight of tender, 36 tons. $\mu_1 = 1/5$. Weight of vehicles behind tender, 284 tons. Then the maximum retardation which the brakes on the coupled wheels can produce on the engine calculated from equation (15) is

$$\dot{v} = a = \frac{32\,(46 - 16)}{46 \times 5} = 4 \cdot 2 \text{ feet per second per second.}$$

The corresponding retardation produced by the tender and train brakes on the tender and train calculated from (7) is

$$\dot{v} = g\mu = 6 \cdot 4 \text{ feet per second per second.}$$

Hence from (3)

$$T = \frac{46 \times 320\,(6 \cdot 4 - 4 \cdot 2)}{g\,\{46 + 320\}} = 2 \cdot 8 \text{ tons,}$$

and $c$ the common retardation is from (4)

$$c = \frac{46 \times 4 \cdot 2 + 320 \times 6 \cdot 4}{366} = 6 \cdot 1 \text{ feet per second per second.}$$

It should be understood that in practice these maximum retardations are not produced, but it will be seen from equation (3) that, since the tension on the draw-bar depends upon the difference of the retardations $a$ and $b$, a large tension may be produced on the draw-bar when the actual values of the retardations are small.

## 9. Characteristic Dynamical Diagram for a Train stopping from a speed of 60 miles per hour.

The principles underlying the practice of braking are generally illustrated by the diagram, Figs. 4 and 5 (page 910). The train is

assumed to be made up of similar wheel-elements similarly braked. The diagram is drawn for one wheel-element, since the characteristics of the motion of one wheel-element during a stop are identical with the characteristics of the motion of a train composed of a number of similar wheel-elements. The weight of the element is assumed to be 4 tons, and the moment of inertia of the wheel and half axle belonging to it is taken to be $0 \cdot 86$ ton-foot$^2$ units.

Consider first the exhaustion of the translational energy of the element. Part of the energy is exhausted against the resistance due to the motion of the element along the rail and to the resistance produced by the application of the brake-blocks to the wheel, and the motion of the element is maintained against these resistances by means of energy withdrawn from the kinetic energy of the element. Energy is also dissipated in overcoming the resistance of the gradient. It is assumed that the wheel will slip when the sum of the train-resistance and the brake-block resistance reduced to the rim of the wheel is equal to 1 ton. This corresponds to a value of $\mu_1$, between the wheel and the rail of $0 \cdot 25$. Curve 1 is drawn parallel to the speed axis in Fig. 5 to represent this resistance. In doing so, the assumption is made that, when the resistance at the rail reaches this value, the wheel will slip whatever be the speed. Further, let it be assumed that the sum of the frictional resistances reduced to the wheel-rim never exceeds three-quarters of this value, in order to allow a margin against slipping. Curve 2 is drawn to represent this value, and it is assumed to increase gradually from $\frac{1}{10}$ of a ton at the beginning of the stop to $\frac{3}{4}$ of a ton, and then to decrease gradually to $\frac{1}{10}$ of a ton at the actual moment of stopping. The form of the curve is of course quite arbitrary, and depends upon the way in which the pressure between the brake-blocks and the tyres is regulated by the driver.

The chain-dotted lines near the axis show the part contributed by the normal train-resistance and the gradient resistance, from which it will be seen that in comparison with the resistance produced by the brake-blocks, their influence on the stop is negligible, except in exceptional cases, such for instance as when

Figs. 4 and 5.

*Characteristic Dynamical Diagram for a Stop from a Speed of* 60 *miles per hour Every wheel is similarly braked.*

Fig. 4.

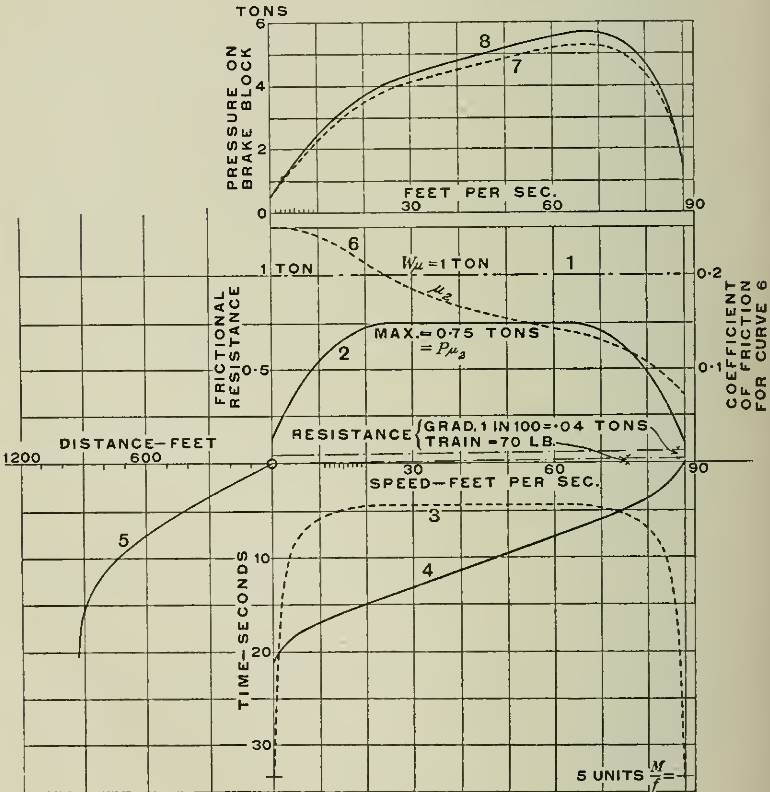



Fig. 5.

the gradient is steep, and the number of braked wheels in the train relatively few.

With passenger-trains stopping in normal conditions of weather on gradients usually found on a main line, and with a large proportion of the wheels braked, the whole resistance against which the translational energy is exhausted may be regarded as produced by the application of the brake-blocks alone.

Ordinates to Curve 2 may therefore be taken to represent the product of the pressure $P_1$ and the coefficient of friction $\mu_2$. Let $P_1\mu_2 = f$.

The process of finding the time-speed curve and the time-distance curve for the stop is from this point exactly the same as the process explained above in connection with Fig. 1 (pages 882–3), with the difference in detail that in this case it is the actual mass of the wheel-element which is to be used, without any increase to allow for the rotatory energy in the wheel. This is considered separately below.

The mass is thus $4/32 = 0{\cdot}125$ unit. Curve 3 is plotted from $M/f = 0{\cdot}125/f$, and $f$ is scaled from Curve 2. This curve integrated with regard to the speed gives the time-speed Curve 4, and Curve 4 integrated with regard to the time gives the time-distance Curve 5.

It will be seen from these two curves that the train is stopped from 60 miles per hour in 21 seconds, and during that period it passes over a distance of 925 feet.

Curves 1 to 5 constitute the dynamical diagram for the stop. In order to produce the frictional resistance represented by Curve 2, the pressure which is applied to the brake-blocks must be varied as the coefficient of friction $\mu_2$ varies.

Curve 6 shows the value of the coefficient of friction between cast-iron brake-blocks and steel tyres plotted as a function of the speed from data given in Table 2 (page 912). These data must not be taken as generally applicable to all cases, but rather as showing the order of variation which may be expected. The curve itself represents values of $\mu_2$ actually found for cast-iron blocks on steel tyres during experiments carried out with extreme skill and care,

with specially constructed apparatus on the London, Brighton and South Coast Railway.

TABLE 2.

*The value of the coefficient of friction between cast-iron brake-blocks and steel tyres in terms of the speed and the time. Quoted from Captain Douglas Galton's Paper on the "Effect of Railway Brakes," Proc. Inst. Mech. Engineers, Oct.* 1878, *page* 599.

| Dynamic Friction. Cast-iron on Steel. | Speed in | | Value of the coefficient of friction, $\mu_2$. | | | | |
|---|---|---|---|---|---|---|---|
| | Ft. per second. | Miles per hour. | At beginning of expts. | After 5 secs. | 10 secs. | 15 secs. | 20 secs. |
| Just before coming to rest | 1 to 3 | 2/3 to 2 | 0·25 | — | — | — | — |
| When moving at | 10 | 6·8 | 0·242 | — | — | — | — |
| " " | 20 | 13·6 | 0·213 | 0·193 | — | — | — |
| " " | 25 | 17·0 | 0·205 | 0·157 | — | 0·110 | — |
| " " | 30 | 20·4 | 0·182 | 0·152 | 0·133 | 0·116 | 0·099 |
| " " | 40 | 27·3 | 0·171 | 0·130 | 0·119 | 0·081 | 0·072 |
| " " | 45 | 30·7 | 0·163 | 0·107 | 0·099 | — | — |
| " " | 50 | 34·1 | 0·153 | — | — | — | — |
| " " | 55 | 37·5 | 0·152 | 0·096 | 0·083 | 0·069 | — |
| " " | 60 | 40·9 | 0·144 | 0·093 | – | — | — |
| " " | 70 | 47·7 | 0·132 | 0·080 | 0·070 | — | — |
| " " | 80 | 54·5 | 0·106 | — | — | 0·045 | — |
| " " | 88 | 60·0 | 0·072 | 0·063 | 0·058 | — | — |

If any ordinate of Curve 2 which gives $P_1\mu_2$ be divided by the corresponding ordinate of Curve 6 which gives $\mu_2$, the quotient is the value of the pressure $P_1$ which must be applied to the brake-block to actually produce the resistance shown in

Curve 2. Curve 7 has been plotted in this way. Owing to the small value of $\mu_2$ at 60 miles per hour, a considerable pressure is required to skid the wheels at this speed. For example, to produce the skidding resistance of 1 ton would require a pressure of nearly 14 tons on the brake-blocks, but just before stopping, where the coefficient of friction is about 0·25, a pressure of 4 tons would skid the wheels. In the example, a pressure is assumed to be applied to one brake-block. If there are two brake-blocks, the same frictional resistance is produced by half the pressure applied to each block.

The values of the pressure shown in Curve 7 are those required to produce the friction necessary to abstract the translational energy from the wheel-element. The pressure must actually be increased as shown by Curve 8, the extra pressures being required to produce a frictional couple equal and opposite to the couple corresponding to the angular retardation of the wheel. With no slipping, the angular retardation is calculated from the linear retardation.

Thus consider the point on the time-speed Curve 4 corresponding to 54 feet per second. At this speed the retardation is uniform and equal to 6 feet per second per second. The radius of the wheel is 1·89 foot (3 feet $7\frac{1}{2}$ inches diameter). The corresponding angular retardation is $6/1{\cdot}89$ radians per second per second. The moment of inertia of the wheel is $0{\cdot}86/g$ in dynamical units $= 0{\cdot}0268$. Hence the force of the couple required to produce this angular retardation is found from

$$P_2\mu_2 = \frac{6 \times 0{\cdot}0268}{1{\cdot}89 \times 1{\cdot}89} = 0{\cdot}045.$$

From Curve 6 the corresponding value of $\mu_2$ is 0·15. Hence the increase of pressure $P_1$ is $0{\cdot}045/0{\cdot}15 = 0{\cdot}3$ ton. Points on Curve 8 have been calculated in this way, and its ordinate represents the force which must be applied to the brake-blocks to produce a frictional couple against which both the translation and the rotatory energy of motion is dissipated, a couple represented above by B (neglecting the effect of the other small resistances),

and it shows how the pressure must be varied in order to produce the stop represented by the characteristic curves.

## 10. Moments of Inertia of typical pairs of Wheels and Axles.

As the value of the moment of inertia I of a pair of wheels enters into several calculations besides those considered above, typical cases have been taken and the values of I accurately calculated. Mr. Pickersgill, Locomotive Superintendent of the Great North of Scotland Railway, has kindly supplied the data required for this purpose. Mr. Deuchar, assistant in the Civil and Mechanical Department of the City Guilds Engineering College (formerly Central Technical College), made the detailed calculations. The method adopted is to divide the wheel into a series of concentric rings, and then to calculate the moment of each ring separately.

The following Table is useful for reference:—

### TABLE 3.

*Moments of inertia; radii of gyration; total weight; and the ratio of the square of the radius of gyration to the square of the radius of the wheel, in four typical cases.*

| Type of wheel. | Total weight of 2 wheels and 1 axle. | $k^2$. | Moment of Inertia. | $k^2/r^2$. |
|---|---|---|---|---|
| | Lb. | Ft.$^2$ | Lb.-ft.$^2$ | |
| 1 pair of driving-wheels 6 ft. 1 in. diameter, connected by a crank-axle. . . . . . . | 8,473 | 4·1 | 34,133 | 0·444 |
| 1 pair of trailing-wheels connected by a straight axle . . . | 6,723 | 4·52 | 30,389 | 0·49 |
| 1 pair of 3 ft. 9½ in. bogie-wheels and axle . . . . . . | 3,080 | 2 | 6,164 | 0·555 |
| 1 pair of 3 ft. 9½ in. wood-centred carriage-wheels and axle . . | 2,356 | 1·63 | 3,840 | 0·453 |

The following rule for estimating the moment of inertia is based on the assumption that the engine-wheel is composed of

tyre, wheel-rim and Mansell rings and $n$ spokes, each of uniform section, and each of length $\rho$ feet, the distance from the centre of gravity of a radial section through the rim to the centre of the wheel. With this assumption, the spokes would be crowded out at the centre, and the material may be imagined to spread laterally, thus making some allowance for the wheel-boss.

Let A be the area in square inches of a radial section taken through the tyre, the wheel-rim, and the Mansell rings. Then allowing 0·28 lb. per cubic inch of material, the moment of inertia of the rim corresponding to this section is with sufficient approximation

$$0{\cdot}28 \times 12 \times 2\,\pi\rho^3 \mathrm{A} \text{ lb.-ft.}^2 \text{ units} = 21\mathrm{A}\rho^3.$$

And the moment of inertia of one spoke, whose mean section is $a$ square inch, about the wheel centre, is

$$0{\cdot}28 \times 12\, a\rho(\rho^2/4 + \rho^2/12) = 1{\cdot}12\, a\rho^3.$$

The moment of inertia of one wheel with $n$ spokes is therefore

$$\rho^3(21\mathrm{A} + 1{\cdot}12\, na) \text{ lb.-ft.}^2 \text{ units.}$$

For a pair of wheels and the axle it may be taken double this. Before using the value so obtained in dynamical calculations, divide by $g$.

Similarly for a wood-centred carriage-wheel, the moment of inertia of the rim is $21\mathrm{A}\rho^3$. This is about 85 per cent. of the total, so that the moment of inertia for one wheel is approximately

$$49{\cdot}4\ \mathrm{A}\rho^3 \text{ lb.-ft.}^2 \text{ units}$$

for a pair of wheels and an axle.

Thus finally rounding the figures off:—

Moment of inertia of a pair of wheels and axle with $n$ spokes may be approximately calculated from

$$\mathrm{I} = 2\rho^3(20\mathrm{A} + na) \text{ lb.-ft.}^2 \text{ units.}$$

Moment of inertia of a pair of wood-centred carriage-wheels and axle may be approximately calculated from

$$\mathrm{I} = 50\mathrm{A}\rho^3.$$

By way of illustration in the case of the trailing-wheel of Table 3 (page 914)—

| | |
|---|---|
| A the area of the cross-section of the rim . . . . . | = 26·6 square inches. |
| $\rho$ the distance of the centre of gravity of this section from the centre of the wheel . . . | = 2·85 feet. |
| $a$ the mean area of a spoke . . | = 5·8 square inches. |
| $n$ the number of spokes . . | = 20. |

Then, I = 30,000. Comparing this with 30,389, the accurate figure, it will be seen that the error is about 1·3 per cent.

This expression makes no allowance for a balance-weight, but where there is one, its moment of inertia can be separately calculated and added to the sum found from the formula.

Adding, therefore, 400 to I to allow for the balance-weight, I = 30,400, and the error is less than 1 per cent.

In the case of the wood-centred carriage-wheel—

| | |
|---|---|
| A . . . . | = 15·55 square inches. |
| $\rho$ . . . . | = 1·7 foot. |
| I . . . . | = 3,880 which, compared with |

3,840, the accurate figure, shows an error of less than 1 per cent.

The Paper is illustrated by 5 Figs. in the letterpress.

*Discussion.*

The President called the attention of members to a new departure made by the Institution in submitting this evening a second Paper for discussion in writing only. The Paper was on "Theory and Experiment in the Flow of Steam through Nozzles," by Professor J. B. Henderson, D.Sc., of the Royal Naval College, Greenwich, and it was hoped the members would read the Paper and forward any communications they had to make upon it to the Institution.

In proposing a hearty vote of thanks to the author for his extremely interesting and valuable Paper, the President said he had nothing to say upon the subject except that it was a splendid illustration of the value of the graphic method in dealing with, and making the ordinary man understand, very difficult mathematical problems. He did not know whether that fact was capable of explanation, but he would much like to know whether there was any physical explanation for the fact that it was easier for most minds to understand the graphic method than it was to understand a complicated analytical equation. He remembered once talking to an old master of his upon the question, and his view was that mathematics in the ordinary sense was not really properly taught to the young and inexperienced mind, but his (the President's) own idea was that the real cause went deeper, that it had something to do with the peculiar reasoning faculty of the brain, and that it was only very exceptional minds that could follow abstruse complicated mathematical formulæ and thoroughly understand them, although when the subject was presented in such a lucid way as the author had presented it that evening it seemed to be quite easy. That was, however, where the master mind showed its power, by making the matter easily understood.

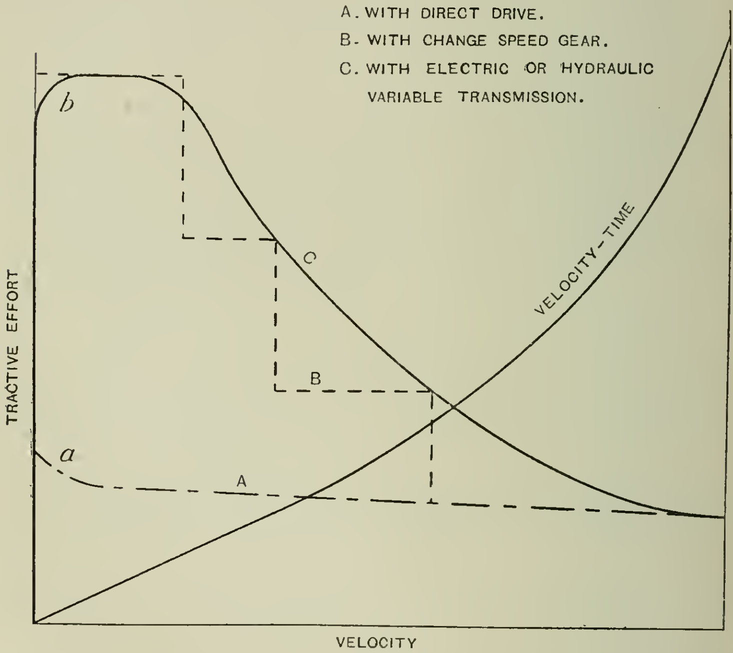
The vote of thanks was carried unanimously.

Dr. H. S. Hele-Shaw (Member of Council) said that, after the remarks of the President, it was not necessary for him to enlarge upon the value of the methods which the author had brought

(Dr. H. S. Hele-Shaw.)

forward that evening, or upon the extremely clear manner in which he had dealt with an abstruse subject in connection with which there was so much that was new and original. He did not see how it was possible to discuss the details of the curves, because he did not think anyone could dispute the accuracy of anything the

Fig. 6.—*Characteristic Curves for Internal-Combustion Engine Locomotives.*




author had said. He would, however, draw attention to the fact that, apart from the steam-engine and the electric railway, there was what might be termed a new phase of locomotive engineering brought about by the introduction of the internal-combustion motor.

It had struck him that it might be useful if he applied the author's method to this aspect of the question of acceleration. He had not done that in quite the same way as the author, because he had not

access to the same instruments or the same data, but he had treated it in a manner which he would very briefly explain. Before doing so, he would draw attention to Fig. 6. It would be noticed that it contained a lower nearly horizontal line A, which line represented the tractive-effort curve of an internal-combustion engine with direct drive on the wheels. He had taken the particular case of a Diesel engine of about 1,000 h.p. which was now working on the Continent, built by a Swiss firm. The members would notice there was a certain disadvantage with the internal-combustion engine on direct drive, so far as the initial tractive effort was concerned. Steam had an advantage which was unique, an advantage which could be gained, as the author had pointed out, in a way by using the central station of an electric railway; but with the internal-combustion engine those advantages were not obtainable, unless something else was introduced. The end of the line, it would be noticed, had a slight upward turn shown at *a*; that represented more or less the starting under compressed air, because the particular engine contained a large number of cylinders of compressed air and the starting was got by employing compressed air. It was possible to make the line rise up to the top of the tractive effort on the diagram, but only by carrying a large number of compressed air cylinders. It did not follow, however, that such a locomotive was not of extreme value, for where the particular locomotive was employed, in a flat country where long distances were run without the necessity of continually starting, it was perhaps an admirable form of motor. But supposing it was desired to accelerate rapidly, there was another way of doing it, represented by the dotted step line B on the diagram which represented the tractive-effort curve in the case of an ordinary gear-box. The problem had been solved beautifully for the common road by the motor-car, which vehicle could start with a considerable load and rapidly accelerate as shown by the step lines of the curve. The triangular spaces, however, represented lost work. Again, taking a vehicle, say with four speeds, to run backwards it was necessary to have four speeds in reverse, and that added another difficulty to the problem. A comparatively small rail-car would weigh 40 tons; a heavy road vehicle would not

(Dr. H. S. Hele-Shaw.)
weigh more than 7 tons (3 tons the vehicle and 4 tons the load), and there were great difficulties in step-by-step motion for rail locomotives.

Coming to the possibility of using either electric or hydraulic transmission, some day engineers would be discussing the relative merits of those two systems, and there might be a very keen discussion on the subject in the Institution. But he wished to make no distinction that evening, and as he noticed many of his electrical friends present he would say that it was possible to obtain equal results, as far as the diagram went, with either system, although he had actually obtained the figure shown by means of a hydraulic transmission.

He had not access to the same kind of instrument as the author had had, and therefore he might explain how that diagram was obtained. He borrowed an accelerometer from the District Railway Co., who kindly lent it for the purpose, and obtained the acceleration curve; the instrument was nothing more than a pendulum-bob writing its story. The pendulum-bob (the Wimperis accelerometer) wrote the curves, and from the curves Mr. Beecham worked out the diagram. It would be seen that Mr. Beecham and himself had produced the same diagram as the tractive-effort curve of the author, although they did not obtain it on the author's basis, having had to go a different way to work by using a time base. Having obtained a curve on the time base, from that he obtained the tractive-effort curve on a velocity base, *C*, Fig. 6, which turned out to be a very similar curve to the author's tractive-effort curve, and reproduced almost exactly the advantage in starting of a steam-engine, or indeed of any other starting which gave a continuous variation in effort as the speed went up. There was a point of interest in the curve which did not run straight horizontally at the commencement as it did in the author's diagram, but curved at *b* in a way which showed that the control did put on the full effort at once, and therefore the maximum tractive effort was not immediately obtained. He only brought the matter forward in detail to show that the same result could be obtained by using an infinitely variable transmission gear with

an internal-combustion engine, as with the expansion effect of a steam-engine.

In conclusion, he would remark that there might be taken to be three aspects of the problem, namely, (I) Steam which had superlative advantages in a country like Great Britain with abundance of coal and moderate distances to be traversed. He could not quite believe that any other form of general locomotion was (as long as our coal lasted) ever going to supersede it, at any rate in this country. (II) For suburban traffic, in great centres of population, the steam-engine was already being superseded, as it was possible to draw upon a central electric station and get rapid acceleration. It would have been impossible to raise the traffic of the District Railway as it had been raised, from thirty million to more than seventy million passengers per annum, unless the line had been electrified and the consequent rapid acceleration had been possible, by drawing on the central station and so getting the start. Lastly, there was (III) the internal-combustion engine. For countries such as Australia and South Africa, they were already beginning seriously to consider the internal-combustion engine, owing to the fact that the water problem was a difficult one and coal was not readily accessible. Even putting aside the question of long distance, there was the possibility of suburban traffic for small towns, where railways were building up passenger traffic for short distances. At present it did not seem to be possible to electrify such lines, which thus opened a future for the internal-combustion engine with some kind of satisfactory variable transmission.

Mr. Walter Longland said the author, in writing his Paper, had conferred a great boon on locomotive engineers, because in future they would be able to design locomotives with much more confidence than they had ever had before. It might be of service, therefore, to point out an extreme important use of one of the diagrams. He referred to the diagram, Fig. 1 (pages 882–3); the curves numbered 2, 3 and 4, for a given weight of train and average gradient, would always be the same, but curve No. 1

(Mr. Walter Longland.)
depended upon the engine. The horizontal portion of that curve practically depended upon the capacity of the cylinders, while the curved portion practically depended upon the steam capabilities of the boiler. It would be easily seen that by altering these portions of the curve—say by increasing the capacity of the cylinders or by giving a greater steaming capacity to the boiler, or even altering both at the same time—and then taking the difference between curves Nos. 1 and 4, it would be possible to determine how the particular alteration would affect the time required to run a certain distance, as, for example, the distance between two stations. Locomotive engineers had never been able in the past to do this with certainty.

There was one suggestion he would like to make about this diagram. The author had plotted curve No. 4 for the starting gradient, and assumed it to be constant for the whole run. Usually the gradient was far from being constant; in applying this diagram, Fig. 1, therefore, it would appear wiser to plot curve No. 4 from what might be called a "weighed" gradient; by this he meant an average gradient which took into account both the rise and the length of the actual gradients on the particular section which was being dealt with, or perhaps, better still, the curve No. 4 might be plotted from the actual gradients.

When he read the Paper he was extremely interested in seeing the application of the diagrams to actual dynamometer-car records, because some twelve years ago he himself worked on some hundreds of yards of such records for the Great Western Railway Co. The author claimed for his method of using the diagrams that it smoothed out accidental errors; that was perfectly true, but it also smoothed out changes of velocity which were not accidental but which were very important. On the previous evening he had looked up a rough note-book in which he had worked out some of the results obtained from the records mentioned above, and there he found some figures which brought this out very clearly. They referred to a train, weighing 667 tons, which was running at a speed of about 13 miles per hour up a gradient of 1 in 209; the distance was half a mile. The figures he was about to give were

the forces per ton required to accelerate the train; these of course represented the acceleration, for force was the product of mass into acceleration and the mass was constant. The gradient was divided into four equal parts, each of which was therefore an eighth of a mile in length. For the first eighth of a mile the force required to accelerate the train was 1·8 lb. per ton, for the second it was 1·6 lb. per ton, for the third it was 1·3 lb. per ton, and the last it was nothing. Thus through a short distance of half a mile the speed of the train varied considerably. The author's velocity-curve could not show these variations. It might be said that if it were plotted to a large enough scale it would show them; this was true, but if, for example, the journey were one hundred miles the author's method would require, for the scale such as the Great Western Railway used at the time that he worked for it, a piece of paper 300 feet long by about as many broad. He would not go into details, but the average speeds at every 2 seconds were found and plotted on the records; and a smooth curve was drawn through these points. It was tedious work and often took some weeks to do; it gave, however, a means of finding the actual resistance of the train very much more accurately than the author's method did, because the changing acceleration could be found for every position of the train. To further emphasize this point, he gave the following figures; they were all obtained while the engine was on one gradient. In the first 31 seconds the change of speed was half a mile per hour, in the next 30½ seconds the change was a quarter of a mile per hour, in the next 29½ seconds it was 1 mile per hour, in the next 28 seconds it was 0·2 mile per hour, and in the next 27 seconds it was 0·6 mile per hour. These figures showed that there were continual changes of speed, which were smoothed out by a curve such as the author used. Yet that these changes must be taking place was perfectly clear, when it was remembered that the horse-power developed by the locomotive was very approximately constant, and that the train was either going up and down hill or on portions on hills of different slopes practically the whole time. To find train resistances, it was necessary to take into consideration the actual, not the average, change of speed.

(Mr. Walter Longland.)

With regard to diagram, Fig. 4 (page 910), it was very unfortunate that the speeds could not be obtained when their values were less than 40 feet per second. He supposed that this was due to the record being only 1 foot per mile run. About fifteen years ago the records obtained from the dynamometer-car then in use on the Great Western Railway were 3 feet per mile run. From these it was possible to obtain the speeds down to practically zero. It was a great pity, he thought, that the Great Western Railway Co. had changed the speed of the movement of the paper. Notwithstanding the criticisms which he had advanced, he felt sure that the Paper would be greatly appreciated by, and be of great value to, locomotive engineers, and he could only say that he was extremely sorry it was not written fifteen years ago.

Dr. William H. Maw (Past-President) said he shared entirely the views which the President and Dr. Hele-Shaw had expressed as to the value of Professor Dalby's Paper. The author had treated the whole subject most admirably, and the Paper would be a valuable addition to the Proceedings of the Institution. He himself only wished to speak on one small point. The author had referred to the allowance for the rotational acceleration of the wheels of rolling-stock. Some thirty-eight years ago he himself had to make a number of experiments on the effect of brakes, and he had to take into account the acceleration—or retardation in his case—to which the author had referred, but he treated the matter in rather a different way, though arriving at practically the same result. The rotational speed of the rolling surface of a tyre was of course equal to the velocity of the train. The travel of the flange would be rather higher than the velocity of the train, and the inner parts of the tyre would have a lesser velocity. The spokes of the wheels and the boss travelled of course at still lower velocities. But he found that practically the effect of these lower velocities was compensated by considering the whole section of the tyre to be travelling at the speed of the train. He made calculations for a number of different types of carriage wheels, and he found that, by simply taking the weight of the tyre and adding it to the weight

of the train, it was possible to get the allowance the author had spoken of. All the centre parts of the wheels were ignored, and the whole section of the tyres was treated as travelling at the surface speed. Speaking from memory, he believed he found the allowance necessary in those days to be about 9 per cent. of the weight of the train. The author had arrived at 7 per cent. Those two results really corroborated each other. At the time to which he had referred, the load per wheel was much less than it was now, and consequently the effect of the tyre acceleration was greater in proportion to the whole weight of the train than at present. In the old four-wheel coaches and six-wheel coaches then used the load per wheel was very substantially less than now. He believed that, if his old method of dealing with this matter was applied to present rolling-stock, the correction required would be likely to come out to about what the author had mentioned, namely, 7 per cent. of the weight of the train.

There was another matter he might mention, although it was, perhaps, not strictly connected with the Paper. It was a certain fact which much facilitated calculations connected with the questions the author had brought forward. A body falling freely from a height of 30 feet would acquire a speed of almost exactly 30 miles an hour at the end of its fall. It therefore followed that, by squaring the velocity in miles per hour of a train and dividing by 30, it was possible to obtain the height from which the train would have to fall to acquire that speed. That was of great assistance in making rapid calculations as to the amount of retardation required to stop in a certain distance or to accelerate in a certain distance. For instance, $\frac{S^2}{30} = H$, H being the height the train had to fall. Thus, for a speed of 30 miles per hour, the value of H was $\frac{30^2}{30} = 30$ feet; for 60 miles per hour, $\frac{60^2}{30} = 120$ feet; and so on.

Multiplying the weight of the train by the height it had to fall gave the number of foot-tons required for the acceleration, so that a 200-ton train required the expenditure of $200 \times 30 = 6{,}000$ foot-tons of work to impart to it a speed of 30 miles per hour.

(Dr. William H. Maw.)

If this acceleration had to be acquired in a distance of, say, 1,000 feet, all that was necessary was to divide this 6,000 by 1,000, giving $\left(\frac{6,000}{1,000} = \right)$ 6 tons mean draw-bar pull to affect the acceleration required.

In that way it was possible to make rapid estimates of the draw-bar pull required to produce given accelerations—or brake resistance to produce a given retardation—within certain distances. Of course, such calculations gave results independent of the effects of friction, or air resistance.

Mr. Henry Fowler said he felt indebted to the author for the way in which he had referred to Mr. Aspinall's experiments on train resistances, as one of those who assisted in the carrying out of the experiments upon which the Paper was based. In the beautiful diagram which the author had put forward, he showed in the resistance-velocity curve the resistance first falling away and then rising fairly rapidly, whereas the curves on Fig. 3 (pages 896–7), based upon the result of the running on the Great Western Railway, showed only a very gradual increase in the train resistance as the speed increased. In the chart and Tables, which the author had referred to as being shown in Mr. Aspinall's Paper in 1901, were to be found all the known formulæ of train resistances from the very commencement of traction on rails until the date of the Paper. It was interesting to remember that some experiments were carried out on this subject in very early days, and he believed that Mr. Edward Wood, who spoke at that Meeting, referred to experiments which were mentioned and which he had carried out 66 years before. The whole of these formulæ, without a single exception, showed that the resistance per ton increased, and did so fairly rapidly, as speed increased, especially at speeds corresponding with those at which observations were made on the Great Western Railway. He did not know whether the author could explain how the results he had given of practical working seemed to be opposed to all previous experiments with a dynamometer-car. He was perfectly aware of the fact that no one had yet succeeded in

producing a "fat" indicated diagram of a locomotive running at high speed.

One point which the author had not touched upon, and a point which, equally with gradients, had a very disturbing effect upon train resistance, was the question of wind, and he had thought that such an important item would have been mentioned in the Paper. He trusted that now, when so much interesting information must have been collected together with regard to wind in connection with the experiments which had led to the successful flying with machines heavier than air, someone, either in discussion or correspondence, would be able to speak upon this, a subject about which so little was definitely known by locomotive engineers. He thought the author would agree with him there was not the slightest doubt that air resistance increased with some power (probably the square) of the velocity, and this rendered the curves based upon the Great Western Railway experiments still more remarkable. It would be noticed that the actual curves themselves, apart from the assumed dotted lines, were, practically speaking, straight, and although he appreciated the fact that the Great Western Railway had a very fine dynamometer-car, yet the time line with 2-second spaces was, in his opinion, altogether too great a time interval for accurate results. It was perfectly simple and easy to get instruments which would give times in fifths of a second with very considerable accuracy, the paper having to travel at much greater speed. The paper used on the Midland travelled 8 feet per mile, although above 20 miles an hour 2 feet of paper per mile was generally used. Even this working of one-fifth of a second was hardly accurate for taking starting accelerations, but instruments were now in use which enable accelerations at low speeds to be readily obtained. He thought the Institution was very much indebted to the author, not only for his Paper, but for the very explicit way he had placed it before the Meeting.

Mr. A. P. Trotter said that those who were familiar with speed curves and acceleration curves must have been surprised at the way in which the author had dissected them into various

(Mr. A. P. Trotter.)

components, components which some engineers, he thought, hardly realized. The author had shown not only how to dissect but how to build up, and the manner in which he had done it would be characterized by some of the older school of professors as "calculus dodging." The members would be extremely grateful, as the President had said, to Professor Dalby for showing how so powerful an instrument as graphics could be used in connection with the subject under discussion. He believed there were some mathematicians who quarrelled with the use of the expression "force of acceleration." In 1906 he himself read a Paper on "Acceleration and Accelerometers" at the Junior Institution of Engineers,* and objection was taken by a mathematician to his use of that expression. It was said that it was at least a hundred years out of date, going back to the time when they used to speak of the *vis inertiæ*, and so forth. It might be a shorter way of saying the force which produced acceleration, but he would like to hear from Professor Dalby whether the term might be used to denote a force.

Acceleration was naturally one of the most important questions of the day in modern locomotion, and, as had been said, was one of the greatest advantages that could be derived from electric traction. There were many who had amused themselves on tedious railway journeys by taking the speed of the train from the window by the milestones, or in the old days, when it was possible to rely on the rails being the same length, by counting the number of rails in a given time. It was easy and very interesting to watch acceleration, and it could be done by a very simple instrument which he himself made about fourteen years ago—an ordinary spirit level bent to a rather shorter curve than usual.† Anyone could make this instrument by bending a glass tube to a radius of about 18 inches, filling with spirit or water, leaving a bubble not more than $\frac{3}{16}$-inch diameter. The scale was graduated by placing the

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* Journal, Vol. xvi, page 255. Reprinted in *Engineering*, 9 March 1906, page 327.

† See *Engineering*, loc. cit.

instrument on known gradients. A slope of 1 in 10 corresponded to an acceleration of one-tenth of $g$, namely, 3·22 feet per second per second. Messrs. Everett, Edgcumbe and Co. made these accelerometers with a curve resembling a cycloid giving an expanded scale in the middle; the diameter of the tube was chosen to make the bubble dead beat. Scales could be supplied graduated in feet per second per second, metres per second per second, or the time taken to reach 30 miles an hour, or gradients, or pull in pounds per ton (about 70 times the feet per second per second).

Such an instrument could not get out of order unless broken; it was subject to no errors and therefore contained no compensations. It would only indicate accelerations in the direction in which the bubble moved. The sensitiveness was a question of length as in a thermometer. The distance between 0 and 2 feet per second per second on the scale of Messrs. Everett, Edgcumbe's pocket instruments was over 1 inch, enabling acceleration to be read easily to one-twentieth of a foot per second per second. By placing it on the window-sill of the train it was possible to see not only the acceleration but every stroke of the piston of a steam-engine as it started, and from that one could deduce, in the way the author had shown, half a dozen different factors including the draw-bar pull per ton, using 69·8 as a factor. Dr. Maw's formulæ reminded him that he had heard it advocated that there should be a law that all motor-cars should have a brake to stop them dead. At 30 miles an hour the effect would be like going into a brick wall at that speed.

Mr. George P. Spooner, referring to section 8 (page 907), said he could corroborate that two unequally braked vehicles produced such severe tension on the draw-gear as to cause rupture. When present at brake trials with a six-coach bogie train running at about 40 miles per hour, the brakes were suddenly applied causing a concertina effect on the vehicles and a parting of the train. On coming to a stand, an examination of the parted train was made, when, to the best of his recollection, of one coupling one of the closing nuts was severed, the screw partly so, and the links much strained; on another the turn-buckle severed and other major

(Mr. George P. Spooner.)

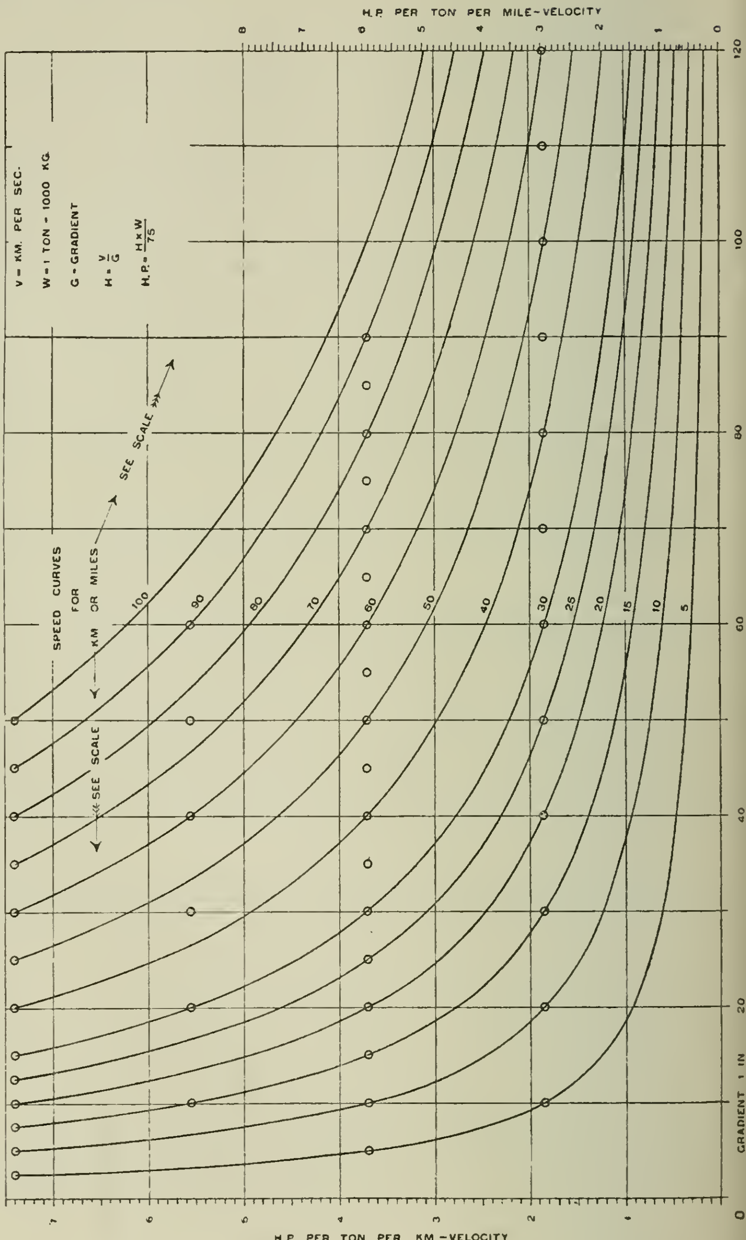


FIG. 7.—Speed Curves.

*To ascertain the Horse-Power per Ton on any gradient for Km.-velocities and for Mile-velocities.*

damage occurred. At that time the rapid-acting valve was not in use.

After the Discussion Mr. Spooner exhibited a diagram slide, Fig. 7, of curves giving the theoretical horse-power required per ton of load for speeds of 5 to 100 kilometres or miles, on gradients of 1 in 5 to 1 in 120, calculated in accordance with the metric formula

$$\text{Horse-power} = \frac{\text{H} \times \text{W}}{75}$$

in which $\text{H} = \frac{\text{V}}{\text{G}}$; G being the grade, H the rise or lift per second due to velocity and gradient, V the velocity of the train, W the weight of 1 ton or 1,000 kilograms and the divisor 75 metre kilograms per second, of which $\eta$ h.p. will be the actual horse-power required. The coefficient $\eta$ would vary with climatic, atmospheric and working conditions, and would be composed of one or more of the following resistances:—Head resistance; side winds; head winds; whether vehicles were loaded or empty; curves; track conditions; and friction. The curves in the diagram could be utilized for velocities either in kilometres or miles.

To ascertain the horse-power per ton on any gradient, for kilometre velocities read from the left of the diagram, and for mile velocities from the right. For instance: 40 curve and gradient 1 in 40, the coincident horse-power on the left gives 3·7 h.p. per ton, while 25 curve read from the right gives the same horse-power, 40 km. being equal to 25 miles.

Professor W. E. Dalby, replying to the discussion, said that the answer to the President's question, why graphical methods were so much easier to follow than analytical methods, would probably be found in the fact that the graphical representation of an equation presented to the eye an infinite series of solutions of the equation in the form of a curve and so enabled a comprehensive conception of the relations to be obtained at a glance.

The "weighed gradient" suggested for curve 4 by Mr. Longland (page 922), though useful in certain cases, would not

(Professor W. E. Dalby.)

give much help in the particular cases where the diagrams would be found most useful, namely, in estimating the motion of the train during the starting and stopping period of a service in which the stops were frequent and the stations close together. The actual gradient should be used in every case of this kind. The author thought that Mr. Longland rather misconceived the purpose of the Paper and the way in which the methods applied in practice. The purpose of the Paper was to exhibit certain fundamental mathematical relations which existed between the various quantities concerned in the motion of a train. An advantage of all graphical methods, not this one in particular, was that accidental errors of observation were smoothed out. If the user of the method applied it so that important changes of velocity were smoothed out also, he had applied it wrongly. There was no suggestion that all the curves were to be derived from the velocity-distance curve. Indeed, it was clearly stated that any curve in the family might be used as a root from which to derive the others. In fact, Dr. Hele-Shaw had used the method by starting with an observed acceleration curve (page 918). Mr. Longland might, for instance, plot a curve from the accelerations quoted from his own note-book, and then obtain a curve which would represent the true acceleration from point to point, because it was certain that the force required to accelerate the train could not change suddenly from 1·8 lb. per ton to 1·6 lb. per ton at the end of the first $\frac{1}{8}$ mile and then change suddenly to 1·3 lb. per ton at the end of the second $\frac{1}{8}$ mile and so on. The values quoted must be average values taken over the intervals of distance mentioned. An acceleration curve plotted from these values would furnish the root curve from which the corresponding velocity curve could be constructed, a curve from which the actual velocity could be inferred with much greater accuracy than from isolated calculations based on the figures quoted.

Mr. Longland advanced as a criticism that to represent a journey of 100 miles long, a piece of paper 300 feet long by 300 feet wide would be required. It was hardly necessary to state that a diagram could be drawn on a double elephant sheet to

represent any part of the journey requiring analysis; that scales might be chosen to suit the conditions of any problem; and that any origin might be used from which to start drawing a diagram. Thus, if the motion up a particular gradient were required, part of the data was the initial conditions of motion at the point selected as the beginning of the diagram, conditions which could quite easily be obtained.

The author quite agreed with Mr. Longland that to find train-resistance it was necessary to take into consideration the *actual*, not the *average*, change of speed. But the figures quoted by Mr. Longland would give *average* rates, not *actual* rates. Thus, Mr. Longland said that in the first 31 seconds the change of speed was half a mile per hour. What could be derived from these figures but the *average* acceleration during the period? The actual acceleration could only be found from a curve.

In reply to Mr. Fowler (page 926), the author said that it would be interesting to plot a diagram obtained from the Midland Railway dynamometer-car because, from the increased scale, data for a velocity curve could be obtained with greater accuracy. With regard to the diagram shown, the author stated that it was put forward to illustrate a method of dealing with dynamometer-car records, which might be useful to locomotive engineers rather than as data concerning train-resistance. The increase in resistance shown, namely, from 11·6 lb. per ton at about 65 feet per second to 13·5 per ton at a little over 100 feet per second was not very great, and did not seem to follow the usual formulæ regarding train-resistance. The information on which it was based, namely, the observed draw-bar pull, gave very definite data about which there could be little question, and it might be that in this particular experiment the wind was helping the train and was tending to reduce the resistance. This might in some measure account for the low resistance at the high speed.

The author had not specifically gone into the question of wind resistance for the reason stated, namely, that the Paper was primarily intended as an illustration of a method which might be used by engineers who had at their command a large amount of

(Professor W. E. Dalby.)
data from which, if they applied the method, results of importance in practice might be obtained with ease. The author hoped that Mr. Fowler would find that the method would prove useful in analysing the large amount of data which he must have at Derby in connection with the trains running on the Midland railway.

Mr. Trotter said (page 928) that the method explained in the Paper would have been called by some of the older school of Professors a method of dodging the calculus. This was probably true. The method was, however, based fundamentally on the principles of the calculus, and part of the analytical machinery was used combined with the graphic method at various points of the process. If one were bound to use the analytical method alone, the problem could not be solved in any useful way, because it was not possible to express in general terms the tractive force as a function of the speed. As explained above, this particular function had at least two discontinuities in it. It was possible to find an analytical function for particular cases, yet the function was in general so complicated that it could hardly be used with advantage in the investigation.

With regard to the question of the suitability of the term "force of acceleration," he (the author) would be inclined to say that it was not quite correct. Strictly speaking, acceleration was the rate of change of velocity, and the force required to produce this rate of change varied as the mass of the body moved. If the mass were negligible, the force was negligible, but the term "force of acceleration" suggested rather that the force depended upon the acceleration only. However, the words used to describe the quantity were quite immaterial, providing the idea attached to the words was correct.

A famous quarrel took place between the leading mathematicians of the seventeenth century, which lasted for forty years, as to the proper way of measuring the force of a body in motion. The measure taken was the product of the mass of the body into its velocity. Leibnitz considered that it should be the product of the mass into the square of the velocity, and all Europe was soon divided between the rival theories. The strangest part of the

controversy, however, was that the same problem solved by mathematicians of opposite opinions always led to the same solution. D'Alembert finally showed in his great treatise on dynamics that the whole dispute was a question of words.

The accelerometer mentioned by Mr. Trotter was an exceedingly interesting little instrument, and he hoped to obtain one soon for experimental purposes.

With regard to Mr. Spooner's remarks (page 929), the author had known many cases of broken draw-bars due to the unequal braking of the trains, and could quite corroborate Mr. Spooner's experience. The author recalled one breakage in particular when travelling on the engine of an express train. The train was approaching a home signal which was expected to fall and did not, with the result that the brakes were suddenly applied. The engine shot forward and left the train behind, the hook between the engine and the tender breaking short with the noise of an explosion. This was due to the unequal braking between the engine and the train, the train having been much more heavily braked than the engine.

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*Communications.*

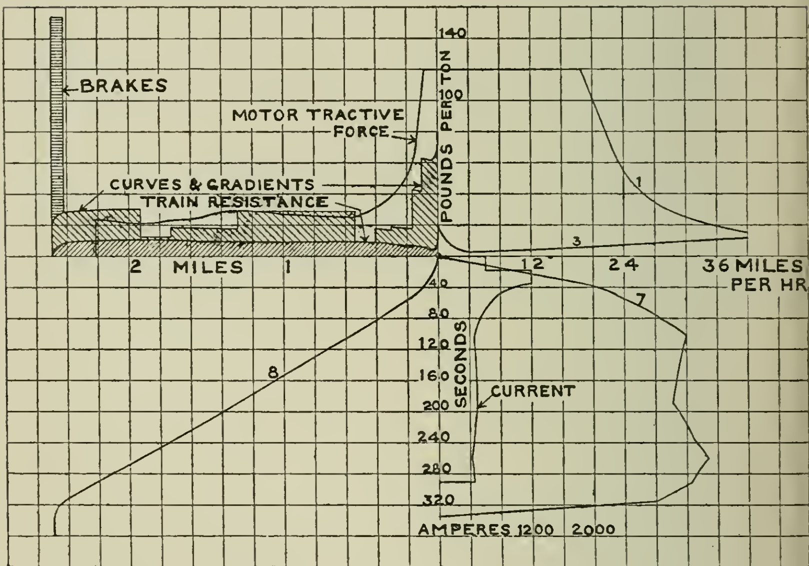
Mr. William Casson wrote that, having during the last twelve years spent a large amount of time in the preliminary calculations for electric railways, he cordially agreed with Mr. Longland (page 924) in wishing that the Paper had been written fifteen years ago. It was of course obvious that the curves for a long journey would cover a very large amount of space, but Professor Dalby had limited his Paper to the accelerating and retarding periods, so that the criticism was hardly apposite. The use of curves connecting speed, current and acceleration with time, and the use of the area of one curve to give the ordinates for the next, and so on, were, of course, quite a familiar study with anyone who had had to do the preliminary work on electric railway undertakings,

(Mr. William Casson.)

Owing to the special suitability of electric traction for short run services, most of the cases to be worked out were of this type, and for these Professor Dalby's method should save a great deal of laborious work.

Since the Paper was read the writer had had to make preliminary calculations for an electric railway, and had used the

Fig. 8.—*Characteristic Curves for* 130-*ton Train equipped with* 4 *d.c. series Motors, each of* 200 *H.P.*




method described in the Paper with good results. The curves for one section of the line, $2\frac{1}{2}$ miles between stops, was shown in Fig. 8. A comparison of these curves with those in Professor Dalby's Paper was interesting as showing the characteristic differences in the performance of a steam locomotive and an electric motor. It would be noticed that the horizontal part of the tractive-force curve No. 1 was much greater in proportion to the limiting speed than in the case of the steam locomotive. This, of course, was where the motor made use of having a power-house behind it.

| From Current Curve. | Kw.-hours. | From Force-distance Curve. | Foot-tons. | Kw.-hours. | Watt-hours per Ton-mile. |
|---|---|---|---|---|---|
| Energy from 3rd rail . . | 25 | Train resistance on level tangent | 6,960 | 5·87 | 18·0 |
| | | Gradients (total rise 83 feet) . | 10,800 | 9·12 | 28·1 |
| | | Additional resistance on curves . | 1,640 | 1·39 | 4·3 |
| | | Brakes . . . . . . | 3,100 | 2·62 | 8·1 |
| | | Total output of motors . . | 22,500 | 19·0 | 58·5 |
| | | From Time-current and Motor characteristic curves:— | | | |
| | | Motor loss . . . . . . | — | 4·4 | 13·5 |
| | | Total input to motors . . | — | 23·4 | 72 |
| | | Controller-resistance loss . . | — | 1·6 | 4·9 |
| | 25 | | — | 25·0 | 76·9 |

Efficiency of Motor Equipment . . 76 per cent. Efficiency of Motors only . . 81 per cent.

(Mr. William Casson.)
This horizontal tractive-force line was maintained by means of a controller, as the speed increased, but when all the controller resistance was cut out and the motors were running on the full voltage of the line, the current fell off as the speed went on increasing, so that the horse-power, instead of remaining constant as in the case of Professor Dalby's steam locomotive, fell off, causing a much more rapid fall in the tractive force.

The curve shown was for a direct-current series motor; with a motor of the shunt type or an alternating-current induction motor, the tractive-effort curve remained horizontal almost up to the maximum running speed, then fell off still more steeply. The advantage of this change of horse-power with speed was that any increase in the resistance of motion of the train due to curves, gradients, or wind which produced a fall in speed, at once produced an increase of horse-power, so that the falling off in speed was less than with a steam locomotive. Once again the power-house came to the rescue. Unhappily for the maker of preliminary calculations, this valuable feature of the electric motor complicated matters very much from his point of view, because the relation of speed to tractive effort at varying loads depended on the design of the particular motor, and curve No. 1 could not be accurately drawn until the motor was decided upon. The method the writer had usually adopted was to work out very roughly the run over an "average section" of the line, that is, a run of average length with an allowance for gradients and curves, and from this to decide on the motor required. The whole line was then worked out in detail, based on the characteristics of the selected motor.

The train-resistance curve looked somewhat high, but the rolling-stock was somewhat light for its bulk, and the number of axles was large. The allowance for curves was 1 lb. per ton for each "degree" of curvature (a curve of 1 degree being 5,730 feet radius). The kinetic energy of rotation was just over 8 per cent. of the total kinetic energy of the train, so that a tractive force of 110 lb. per ton gave an acceleration of 1 mile per hour per second. The weight of the train was 130 tons (of 2,240 lb.) and the voltage on the third rail 550. The braking-force curve was shown as a

horizontal line. This, of course, was not strictly correct, but it correctly showed the average braking force.

The figures arrived at from the curves had been checked by making out a balance sheet for the subdivision of the energy put into the train. This worked out as shown in the Table on page 937.

It might be of interest to show by way of comparison a similar balance sheet for a section of tube railway with stops 0·45 mile apart. There was a falling gradient of 1 in 33 out of each station, and a gradient of 1 in 66 before entering each station. The effect of this could best be shown by comparing with a balance sheet for a similar service without such gradients.

*Watt-hours per Ton-mile with Assisting Gradients.*

| | | | | |
|---|---|---|---|---|
| Energy from 3rd rail | 43 | Resistance loss | 10 | |
| " " gradient | 14 | Motor loss | 2 | |
| | | | | 12 |
| | | Train resistance | | 20 |
| | | Gradient while stopping | 14 | |
| | | Kinetic energy absorbed in brakes | 11 | |
| | | | | 25 |
| | 57 | | | 57 |

*Watt-hours per Ton-mile without Assisting Gradients.*

| | | | | |
|---|---|---|---|---|
| Energy from 3rd rail | 60 | Resistance loss | 12·5 | |
| | | Motor loss | 2·5 | |
| | | | | 15 |
| | | Train resistance | | 20 |
| | | Kinetic energy absorbed in brakes | | 25 |
| | 60 | | | 60 |

The result of the gradients was to reduce the necessary capacity of the motors by 30 per cent. and reduce the brake-block wear by more than 50 per cent. In tube railway work, where dust was specially to be avoided, this was a particularly valuable feature. It would be noticed that, in spite of the much lower speed of the train in the tunnel, the train-resistance figure was actually higher than on the open section, due to the air resistance when running in

(Mr. William Casson.)

a tunnel of very little larger cross section than the train. To show the effect of this, it might be of interest to mention that a train consisting of a single motor-car took just half the current, and ran at the same speed as a train of seven cars of which two were motor-cars, the speed being 27 miles per hour in each case. The motors in this case were those for which the tube section balance sheet was made out, which were about half the rated horse-power of those from which the curves were traced. At a speed of 27 miles per hour the total tractive effort for the single car was 1,000 lb. and that for the seven-car train 2,000 lb.

From the well-known Aspinall formula, the tractive resistances of the car and the train in the open would be respectively 7·2 and 6·65 lb. per ton, giving totals of 166 and 770 lb. There was, therefore, an additional total resistance due to running in the tube which was obviously independent of the weight of the train, but would depend upon its length and sectional area. This additional resistance amounted to 834 lb. for the single car and 1,230 lb. for the train. The speed for the single car and the train being the same, the head and tail resistance would be the same in each case, so that the following result was obtained:—

| | Single Car. | Train. |
|---|---|---|
| Additional head and tail resistance | 768 | 768 |
| ,, side air friction | 66 | 462 |
| Total additional resistance | 834 | 1,230 |

It was interesting to note that 768 lb. was about 8·5 lb. per square foot of cross-sectional area of the train. A water-gauge showed 1·5 inch difference of level between the back and front of the train, corresponding to 7·8 lb. per square foot.

With reference to coefficients of friction for brakes, it might be of interest to describe a machine the writer had fitted up for testing this, and also the relative life of tyre and brake-blocks. It consisted of a shaft driven by a motor with a carriage-wheel fixed on it, to which the brake-block was applied by an ordinary Westinghouse cylinder with triple valve and auxiliary reservoir. It was intended to fit a fly-wheel on the shaft so as to bring the

total kinetic energy of the system up to the kinetic energy to be dealt with by each brake-block in service, but the kinetic energy of the armature, gear, shaft, and carriage-wheel happened to work out exactly right for the purpose.

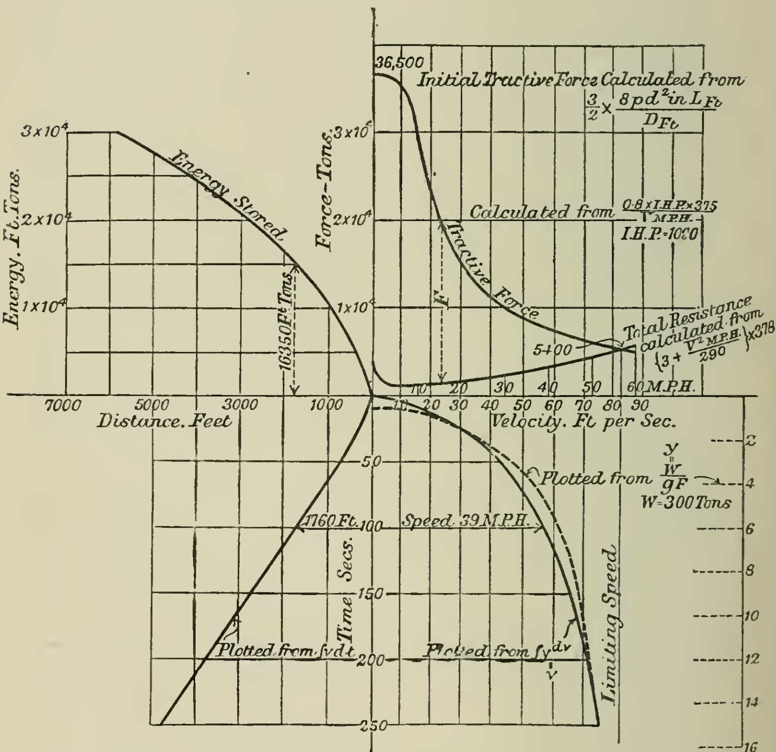
The motor was controlled by being connected to an ordinary controller with the handle fixed on the first notch. Attached to the shaft was a centrifugal governor which, on reaching the particular speed for which it was set, closed an electric circuit and tripped the "dead man's handle" valve of the controller; this let the air out of the train-pipe, and so cut off the current and applied the brake. The train-pipe was supplied with air from a reservoir through an aperture—about $\frac{1}{32}$-inch diameter—which was open all the time. As soon as the brake was applied and the speed began to fall, the "dead man's handle" valve was closed automatically and the train-pipe began to feed up, but owing to the small aperture the pressure did not rise sufficiently to begin releasing the brakes until the wheel had come to rest, when the rise in pressure released the brakes and switched the current on again. The apparatus was thus entirely automatic. For testing the life of brake-blocks and tyres, it was simply set going and left to itself, the speed at which the governor operated being adjusted to about 14 miles an hour, which was about the speed at which the brakes were normally applied. The number of applications of the brake was recorded by a Veeder cyclometer attached to the controller valve-lever, and the wear of the block and tyre measured at intervals as required. For these "life" tests the pressure in the main reservoir varied between 80 and 90 lb. as on the trains, the ordinary compressor and automatic governor being employed. For tests on coefficients of friction a revolution-counter and stop-watch were employed, and a pressure-gauge on the cylinder.

Mr. Bernard M. Jenkin wrote that he thought the Paper was perhaps of even greater value to the engineer who had to deal with electric traction on railways than to the locomotive engineer. These diagrams offered a means of comparing the performance of

(Mr. Bernard M. Jenkin.)
an electric locomotive with a steam locomotive, as was so admirably shown on the diagram, Fig. 10.

In considering the conversion of suburban traffic from steam to electric traction, the very first consideration was the extent by which the service could be accelerated. The performance of the

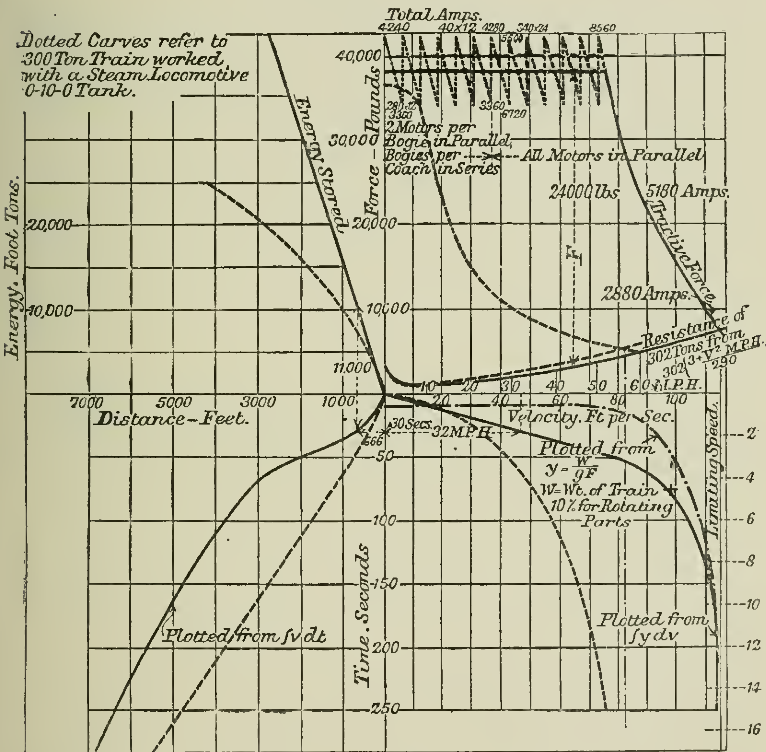
Fig. 9.—*Characteristic Diagram for a* 10-*w.c. Tank-engine weighing* 78 *tons and hauling* 300 *tons.*




existing steam trains could be measured without much difficulty, at least as far as acceleration when starting, average running speed and times from station to station were concerned. But to estimate what the service would be, if converted to electric traction, at once involved calculations and the plotting of curves and diagrams such as described in this Paper.

Fortunately the curves of the performance of an electric motor could be very much more easily and accurately obtained than could the indicated horse-power of a locomotive. The motor could be tested in the shops under different loads and speeds, and a tractive-effort curve obtained corresponding to the i.h.p. curve of the

Fig. 10.—*Characteristic Diagram for an Electric Train weighing* 302 *tons and having* 24 *d.c. Motors.*




locomotive shown in Fig. 1 (pages 882–3). From this curve and the estimated resistance of the train and the known gradients of the line, a set of diagrams could be plotted as described in the Paper. These at once gave an accurate estimate of the time and power that would be taken by a train driven by such motors to do each section of the journey. By drawing these diagrams for different

(Mr. Bernard M. Jenkin.)
sizes and makes of motors, a true and accurate comparison between the motors could be made as to their suitability for the work. At the same time the demand made on the power-house and substations was shown. The full and careful working out of these estimates beforehand was essential to the success of the subsequent working.

The fact that the driver of the train had the full power of the generating station to draw on meant that he would draw on it as he found necessary to keep his schedule time. Unless, therefore, the motors and their gear were properly chosen, he would either be overloading his motors and stripping his gear, or he would be making unnecessarily heavy demands on the power-house and subsequently be wasting power in wearing out his brake-blocks and tyres. The Paper was, therefore, of very real and lasting value to those engineers who had to deal with the electrification of railways, and he was sure they would be much indebted to the author for having given them so complete and useful a tool with which to do their work.

Professor Dalby wrote that he was very glad to see that, even in the short time which had elapsed since the reading of his Paper, Mr. Casson had tried the method and found it useful, and that he had actually employed it to make the preliminary calculations in connection with an electric railway. He was also glad to learn how thoroughly it was appreciated by both Mr. Casson and Mr. Jenkin, gentlemen who had had a wide experience in dealing with problems such as those which the diagram was specially designed to assist in solving.

Characteristic dynamical diagrams were shown at the Meeting with the object of comparing the performance of electric motors with steam locomotives, in connection with the acceleration of trains of the same weight. The curves were exhibited in Figs. 9 and 10 (pages 942–3), from which it would be seen that, as pointed out by Mr. Casson and Mr. Jenkin, it was quite impossible for a steam locomotive ever to compete seriously with an electric motor when it was merely a question of acceleration; with an electric motor, the tractive-force curve could be kept horizontal up to a much

higher speed than was possible in the case of a steam locomotive. The question, therefore, was one of the fundamental difference between the tractive-force curves, and wherever quick acceleration was desired and the service was to be intermittent with a short distance between the stations—conditions which meant that the train was practically always being accelerated or retarded, there being no intermediate period between the stop and the start during which a train could run at any approach to uniformity of speed—then there was no question that the electric motor could produce far greater acceleration than was possible with any locomotive that could be designed or could be put upon the road with the present construction gauge.

# The Institution of Mechanical Engineers.

## PROCEEDINGS.

### November 1912.

An Ordinary General Meeting was held at the Institution on Friday, 22nd November 1912, at Eight o'clock p.m.; Edward B. Ellington, Esq., President, in the Chair.

The Minutes of the previous Meeting were read and confirmed.

The following Papers were read and partly discussed:—

"Vapour-Compression Refrigerating Machines"; by J. Wemyss Anderson, M.Eng., *Member*, Dean of the Faculty of Engineering, the University of Liverpool.

"A Contribution to the Theory of Refrigerating Machines"; by John H. Grindley, D.Sc., *Member*, Principal, Crawford Municipal Technical Institute, of Cork.

The Meeting terminated at a Quarter to Ten o'clock. The attendance was 147 Members and 90 Visitors.

# VAPOUR-COMPRESSION REFRIGERATING MACHINES.*†

By J. WEMYSS ANDERSON, M. Eng., *Member*,
Dean of the Faculty of Engineering, the University of Liverpool.

*Introduction.*—The progress of refrigeration since the last Paper ‡ on the subject was read before this Institution has been enormous. The value of food-stuffs imported into this country, subjected more or less to refrigeration, is now about £130,000,000 per annum, while the application of refrigeration to home produce, to manufactures, to industries, and to the many miscellaneous

* In the Annual Report of the Institution for the year 1906 the Council referred to the question of additional research work, and invited suggestions thereon. In the succeeding Reports (*See* Proceedings 1908, page 86, and 1909, page 218) the Members were informed that inquiries were being made on three subjects with a view to the appointment of Research Committees. Mr. J. Wemyss Anderson was invited to write a preliminary Paper upon the features of Refrigerating Machinery in which further investigation is needed; and the present Paper has therefore been prepared for presentation to the Members, in order that views may be obtained which may be of assistance in the prosecution of the suggested Research.

† In view of some subjects in this Paper having been referred to in the Second French Congress of Refrigeration, September 1912, it is desirable to state that the manuscript of this Paper was received from the author on 2nd February 1912.

‡ "On Refrigerating and Ice-making Machinery and Appliances." By T. B. Lightfoot, Proceedings, I.Mech.E., May 1886, page 201.

purposes, makes the subject one of great importance to scientists and engineers.

The "enormous progress" has, however, been more of a commercial nature than a scientific one; indeed, no branch of mechanical science has received less aid in this country from research or from published accounts of practical progress than that of mechanical refrigeration.

The field for research is very great and includes the machinery proper; the construction, equipment, and insulation of land cold storages; design of ships for carrying refrigerated cargoes; the construction of ammunition magazines (chiefly naval); transportation and storage of perishable produce; the right temperatures for storing various goods; the importance of steady temperatures in fruit carrying and similar cases; the methods of registering temperatures; the condition of the air in cold storages with regard to its temperature, purity (ventilation), humidity, and effect on the goods stored. This list might be extended *ad lib.* if references were made to industries, such as the drying of the air for blast-furnaces, chemical works, soap works, etc.

It is obvious that the actual refrigerating machine demands the first consideration, and of these the principal types in general use are:—

(1) Vapour-compression machines.

(2) Ammonia-absorption machines.

(3) Cold-air machines.

The last named are not now relatively so important as when dealt with by Mr. Lightfoot in 1886.

The ammonia-absorption machine—an important section—can only hope to hold its own against compression machines where waste steam or waste heat in some other form is available.

The first named is, commercially, by far the most important, and on this ground it is proposed to confine this Paper entirely to vapour-compression machines.

*Action of Refrigerating Machines.*—Heat will naturally flow from a warm to a colder body, but if a quantity of matter is to be

made colder than its surroundings, heat must be removed from it by the expenditure of work. This latter action is the function of a refrigerating machine, which, in consequence, is often spoken of as a "heat-pump." A heat-pump is a reversed heat-engine. The latter receives heat at a high temperature, transforms a certain amount of this heat into mechanical work, and rejects the remainder at a lower temperature.

A heat-pump receives heat at a relatively low temperature, and by the expenditure of mechanical work raises it to a higher temperature, that is, sufficiently high to allow the heat to dissipate.* The total heat so dissipated is the heat removed from the cold body together with the heat equivalent of the work expended in removing or "pumping" it out.

If $H_2$ = heat extracted;
$H_1$ = heat dissipated;
$A\,W$ = heat equivalent of work expended;

then $$H_2 + A\,W = H_1$$

and $$A\,W = H_1 - H_2 \quad . \quad . \quad . \quad (1)$$

so that $H_1 - H_2$ = heat equivalent of work expended.

The ratio of the work expended to the heat extracted is taken as a measure of the efficiency of the machine, but this ratio being generally greater than unity, the term "coefficient of performance" (indicated by $\eta$) is used to express the ratio.

Thus for any machine—

$$\text{Coefficient of performance} = \frac{\text{Heat extracted}}{\text{Heat equivalent of work expended}}.$$

Under ideal conditions, from Equation (1)—

$$\eta = \frac{H_2}{H_1 - H_2} \quad . \quad . \quad . \quad (2)$$

* In 1852 the late Lord Kelvin pointed out that the heat from a reversed heat-engine could be economically employed for warming purposes. In the United States refrigerating machines are now employed for cooling large private houses and other buildings in the summer, and warming them in the winter—efficient ventilation being part of the scheme.

Also under ideal conditions the heat $H_2$ would all be taken in at the temperature $\tau_2$, and $H_1$ all discharged at $\tau_1$; hence

$$\frac{H_2}{H_1 - H_2} = \frac{\tau_2}{\tau_1 - \tau_2}$$

and for an ideal machine*—

$$\eta = \frac{\tau_2}{\tau_1 - \tau_2} \quad . \quad . \quad . \quad (3)$$

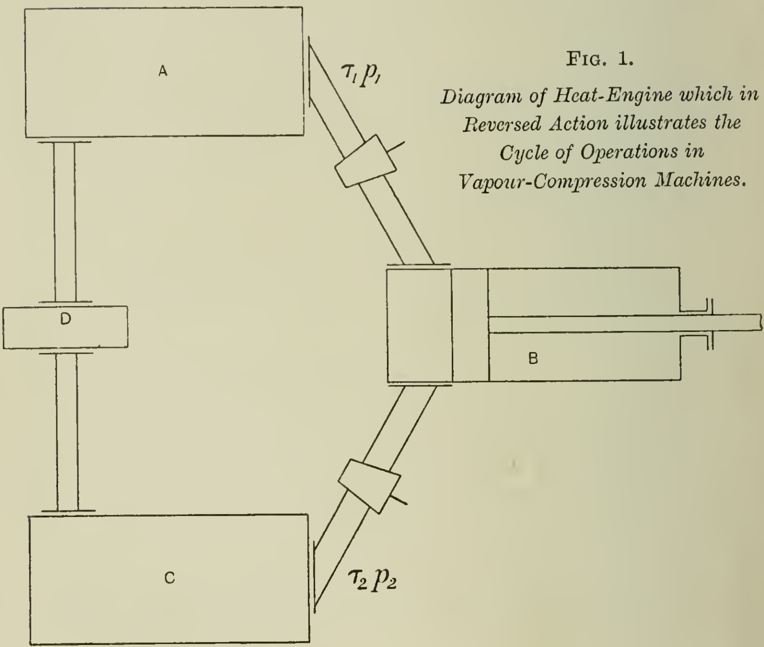



FIG. 1.

*Diagram of Heat-Engine which in Reversed Action illustrates the Cycle of Operations in Vapour-Compression Machines.*

*Cycle of Operations in Vapour-Compression Machines.*—The heat-engine, which in its reversed action most closely resembles the vapour-compression machine, is the reciprocating condensing steam-engine, and it will be an advantage to consider this well-known cycle first.

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* An asterisk indicates that the subject is further treated or demonstrated in an Appendix.

The boiler A, Fig. 1, is supplied with heat and steam generated at $\tau_1$ and $p_1$.

This steam does work in the cylinder B, part of its heat being converted into work, the steam afterwards being condensed (that is, a certain amount of heat is lost or dissipated) in the condenser C at $\tau_2$ and $p_2$. The condensed water is forced back into the boiler by the feed-pump D—this feed-pump calling for an expenditure of work. It is an advantage to keep this feed-water as high in temperature as possible.

In the vapour-compression machine a liquid (refrigerant) is vaporized in the evaporator C, at a pressure $p_2$, by the supply of a quantity of heat from the body (or fluid) to be cooled at temperature $\tau_2$. The vapour so formed is drawn into the compressor B, at a pressure $p_2$, and then, by virtue of work expended, forced into the condenser A, where it is liquefied (generally by the agency of water) at $\tau_1$ and $p_1$. By means of a regulating valve D the liquid at $p_1$ is allowed to pass in regulated quantities into the evaporator C at $p_2$, for re-evaporation, Fig. 1 or Fig. 2 (page 957). This liquid should be kept as low in temperature as possible.

In an ideal vapour machine the regulating valve D would be replaced by an expansion* cylinder, in which the liquid on passing from the higher to the lower pressure would do work. This, it will be seen, would give a closer analogy to a reversed steam-engine, as it would as nearly as possible reverse the action of the feed-pump.

In this cycle of operations there are four thermodynamic losses, namely :—

(*a*) A loss due to the free expansion at the regulating valve; this operation, being non-reversible, renders the entire cycle non-reversible, with a corresponding thermodynamic loss.

(*b*) The refrigerant in the compressor becomes superheated, and this superheat has to be given to the condensing water by a non-reversible operation, the temperatures of the refrigerant and the water being different, resulting in a further thermodynamic loss.

(*c*) To facilitate the transfer of heat between the refrigerant

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* *See* footnote on page 952.

and the condensing water, an appreciable difference of temperature is allowed between them, rendering the cycle further non-reversible.

(*d*) The same applies to the heat transfer between the refrigerant and the brine (or other body to be cooled), supplying the necessary heat for vaporizing the refrigerant in the evaporator.

With regard to the two latter, (*c*) and (*d*), it would be fruitless to discuss the losses further, as the differences of temperature are only of a very moderate nature in practice (10° to 20°) and no greater than actually required for practical purposes.

The loss (*a*) varies with the refrigerant used, and is dealt with in Appendix IV (page 1019), and must be again mentioned in the next section on refrigerants.

Loss (*b*) is the subject of much controversy, and the author has not been able to find conclusive tests on the relative values of "wet" and "dry" compression.* In this country, where so many ammonia and carbonic anhydride machines are used, a compromise may be said to be the general practice.

The vapour is brought back to the compressor in a somewhat wet state (that is, particles of liquid suspended in the vapour), and the amount of superheat is relatively small, making the use of a water-jacket unnecessary.* In the United States as much "cooling effect" is taken out of the vapour as possible, and it goes to the compressor in a dry or super-dry state, and generally becomes highly superheated on compression, the compressors being water-jacketed, although recent practice points to the adoption of the British method of working, which is known in the United States as the "flooded" system.

*Refrigerants.*—Thermodynamically, it does not matter what fluid is employed in a heat-engine or reversed heat-engine, yet certain characteristics, physical properties, and practical considerations limit the choice.

Water-steam has never been seriously challenged as the most suitable fluid for engines working on the Rankine cycle. In the

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* *See* footnote on page 952.

vapour-compression refrigerating machines, however, no one fluid holds an unchallenged position—quite a number are employed, of which three are of outstanding importance, namely, anhydrous ammonia ($NH_3$), carbonic anhydride ($CO_2$) and sulphurous anhydride ($SO_2$).

These are able to stand the temperatures and pressures employed (water, for instance, is not available, as it would freeze at the low temperatures required in refrigeration); their capacity for heat is great and they possess low temperature vaporizing points.

Taking $-4°$ F. and $+86°$ F. as limiting temperatures for the purpose of comparison, the Tables given in Appendix II (pages 1007–9) show that the corresponding pressures for $NH_3$ are 27 lb. per square inch absolute and 170 lb.; for $CO_2$, 288 lb. and 1,038 lb.; and for $SO_2$, 9 lb. and 66 lb. per square inch respectively.

None of these pressures present serious difficulties, but in $CO_2$ machines special care has to be taken, particularly in the compressors, to prevent leakage past the piston and piston-rod and through the stuffing-box or boxes.

In $SO_2$ machines the working pressure in the evaporator, for ice-making and similar low temperature work, is below atmospheric pressure, and there is a danger of air leaking in, which in itself would eventually seriously affect efficient working. This has rather told against $SO_2$ for low temperatures, but for the temperatures required in dairy work the evaporator pressure is above that of the atmosphere; under these and in fact under all conditions, $SO_2$ forms a most efficient refrigerant.*

Ammonia-gas in large quantities is exceedingly noxious, and it is impossible for a man to live in the fumes; hence the Board of Trade will not allow a $NH_3$ machine in the main engine-room of a steamer.

$CO_2$ is not poisonous, but it will not support life. It is much heavier than air, and consequently in a closed space would displace the air necessary for life; hence the Board of Trade will allow only small machines in the main engine-room of steamers.

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* *See* footnote on page 952.

Regarding the thermodynamic losses, that at the regulating valve* tells rather heavily against $CO_2$—$NH_3$ and $SO_2$ being much the same. It is not suggested that an expansion cylinder should be used, but the relative losses must be taken into account when discussing the merits of the three refrigerants. The theoretical loss due to superheating again tells against $CO_2$, while $SO_2$ is rather better than $NH_3$. (*See* Appendix IV, page 1019.)

Taking both these losses into account, it would appear that $SO_2$ is very slightly more efficient than $NH_3$, while $CO_2$ is a very bad third. Here again really reliable comparative tests are wanting, the author having found very conflicting evidence in the published results of actual trials. (*See* Appendix VI, page 1028.)

The practical preference for $NH_3$ over $SO_2$ appears to be in the fact that for all ordinary working temperatures the pressures for $NH_3$ are above the atmosphere, and with any leak the ammonia makes itself felt, while for $SO_2$ air may leak in without indication, as previously pointed out.

The distinct advantage of $CO_2$ is that it is practically inodorous. If reliable tests proved $CO_2$ to be as inefficient as theory points out, the machine would still justify its use on large passenger steamers, but it would be difficult to justify its use on, say, a man-of-war. Such a vessel may live its life without going into action, and supposing an ammonia machine was fitted, it would be quite easy to blow off the charge before coming in touch with the enemy.

Finally, in comparing the three refrigerants, it should be said for a given refrigerating effect* $SO_2$ is the largest machine and $CO_2$ the smallest, but the working pressures (and hence thickness of metals) are in the reverse order, so as regards weight and cost there is very little to choose between the three types of machines.

*Practical Examples of Compression Machines.*—Fig. 2 indicates the principal parts of all vapour-compression machines, viz. :—

| | |
|---|---|
| (1) Regulating valve, D. | (3) Compressor, B. |
| (2) Evaporator, C. | (4) Condenser, A. |

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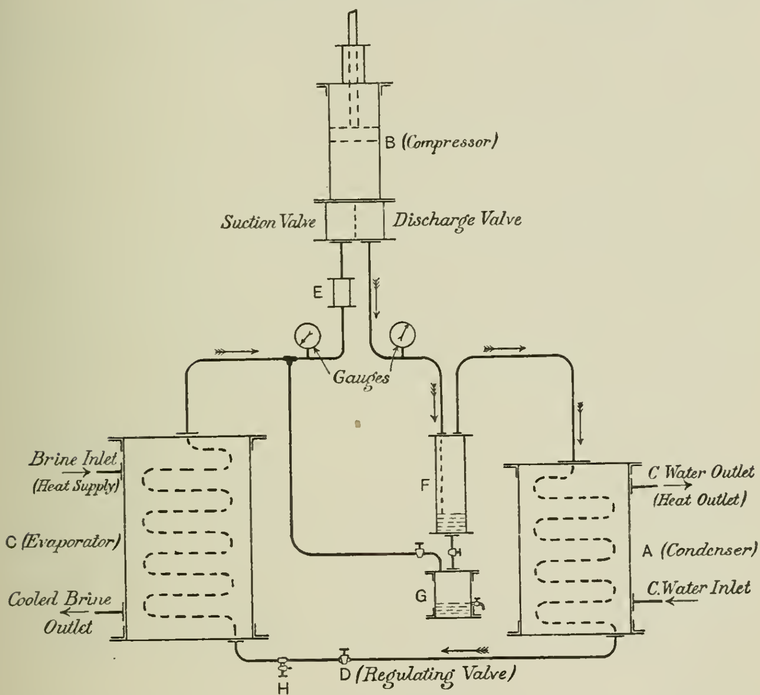
* *See* footnote on page 952.

It is proposed to describe these parts in detail. There are, however, other parts and fittings which are of greater or less importance according to the type of machine to which they are fitted. Fig. 2 indicates some of these. E is a *scale-trap* fitted on the suction-pipe between the evaporator and compressor, and so arranged that the vapour has to pass through a gauze-wire pocket

Fig. 2.—*Principal Parts of Vapour-Compression Machines.*




which effectually traps any scale coming from the long lengths of pipe in use on the machines, and thus prevents scoring of the compressor walls and irregularity in the working of the mushroom valves.

F and G combined form a *rectifier*, the object of which is to catch oil or other fluid passing over from the compressor. F is a cylindrical vessel into which the discharged vapour from the compressor passes by means of an internal pipe, the oil thereby being thrown to the

bottom, the vapour escaping to the condenser by a pipe fitted at the top of the vessel. After a certain period, the length of which would depend on the type and size of machine and on the dimensions of the vessel F, the oil is blown into a second vessel G, after which the connection between F and G is again closed. By a simple arrangement easily followed in the diagram, the vessel is brought under the lowest pressure in the system, and the oil can then be blown off into buckets for re-use.

H indicates a *charging cock*, through which new charges of refrigerant are introduced into the machine. The exact position of this fitting varies with different makers.

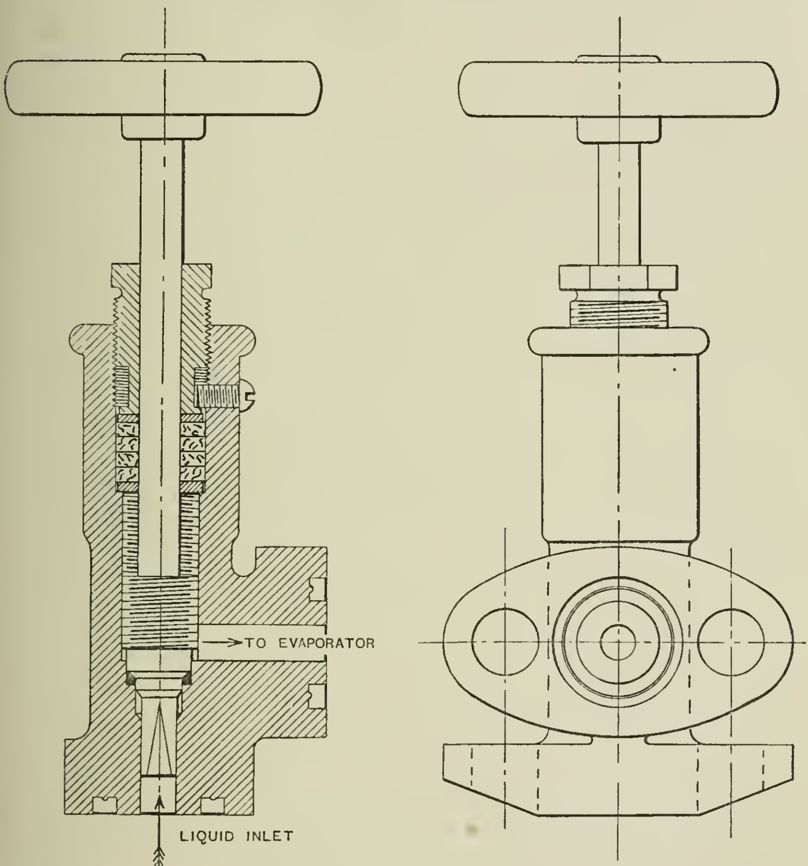
Further, a *liquid receiver* is often fitted between the condenser and regulating valve, while in the United States a "drier," consisting of a cylindrical vessel in which quick-lime is placed, is so arranged as to be capable of being brought into the vapour circuit of ammonia machines, the object being to extract any aqueous vapour that might have worked its way into the circuit and which seriously affects the coefficient of performance. The author considers the question of always fitting *driers* to machines of 10 tons ice-making per day and upwards should receive careful consideration.

*Regulating Valves.*—The amount of liquid refrigerant passing from the condenser to each separate section of the evaporator or evaporators has to be most carefully regulated according to the actual amount of cooling to be performed and the temperature to be carried. The design of the valves necessary for this purpose varies in practice, but in the main they are simple, ingenious and effective.

Fig. 3 shows an ammonia regulating valve which consists essentially of a spindle actuated in the usual way by means of a hand-wheel and plus thread. A part of this spindle is so arranged as to form an ordinary valve, by means of which the liquid can be completely cut off. For regulating purposes the spindle is extended and works as a plug in the body of the valve casting. This plug has a Λ-shaped notch, which not only widens but also deepens from the top downwards. This notch (and the

plus thread) permits of the most minute regulation of the amount of liquid passing. The wheel is often fitted with a finger moving on a dial to indicate the amount the valve is actually open.

Fig. 3.—*Ammonia Expansion Valve.* Half size. (Sterne.)




*Evaporators.*—Although evaporators differ very much in general form, the principle underlying their action and dominating their construction is the same in all machines, namely, the refrigerant is allowed to vaporize in small (1 inch to $2\frac{1}{2}$ inches diameter) tubes or

pipes, the pressure of vaporization being regulated to correspond to the desired temperature. The exceptions to this general rule as to the size of the pipes or vessels used for evaporators are so few as to hardly merit any further mention. Lap-welded wrought-iron pipes electrically welded or jointed together are generally used, but in $CO_2$ machines heavy copper pipes are also employed, both in the evaporator and condenser. It should be noted that copper pipes cannot be used with $NH_3$.

The heat necessary for the vaporization of the refrigerant is supplied by the medium surrounding the pipes—air or brine in the majority of cases, and water, milk, wort, beer, etc., in special machines. This actual cooling constitutes the refrigerating effect of the machine, and in the case of water or milk cooling (say) concludes the work of the machine. Air and brine, on the other hand, are generally but agents, whereby cooling or freezing can be effected at points more or less remote from the machine.

Fig. 4 shows a $CO_2$ vertical evaporator, circular in form, with concentric coils made up of pipes $\frac{1}{1}\frac{5}{6}$-inch bore. This type is common to all vapour-compression machines. As a rule, each coil is separately connected to the inlet header, or manifold at the bottom, and to a corresponding outlet at the top, each coil *either* having the same pitch and different lengths of pipe, *or* the same length with varying pitches. In the present example there are six coils of equal pitch with only two $CO_2$ inlets and two outlets, these four openings being side by side at the bottom. The first inlet passes direct to the innermost or first coil, and on reaching the top is taken to form the second coil, which on reaching the bottom forms the third coil, and this at the top is continued as the fourth, the bottom of which forms the first outlet or suction to the compressor. The second inlet connects direct to the fifth coil, which passes over at the top to the sixth or outer coil, which at the bottom forms the second outlet. The mild-steel shell is made in halves, which are bolted together as indicated. This construction facilitates erection, inspection, and cleaning.

The brine inlet is at the top and the outlet diametrically opposite at the bottom, thus ensuring efficient circulation—a most important matter, considering the difference of temperatures

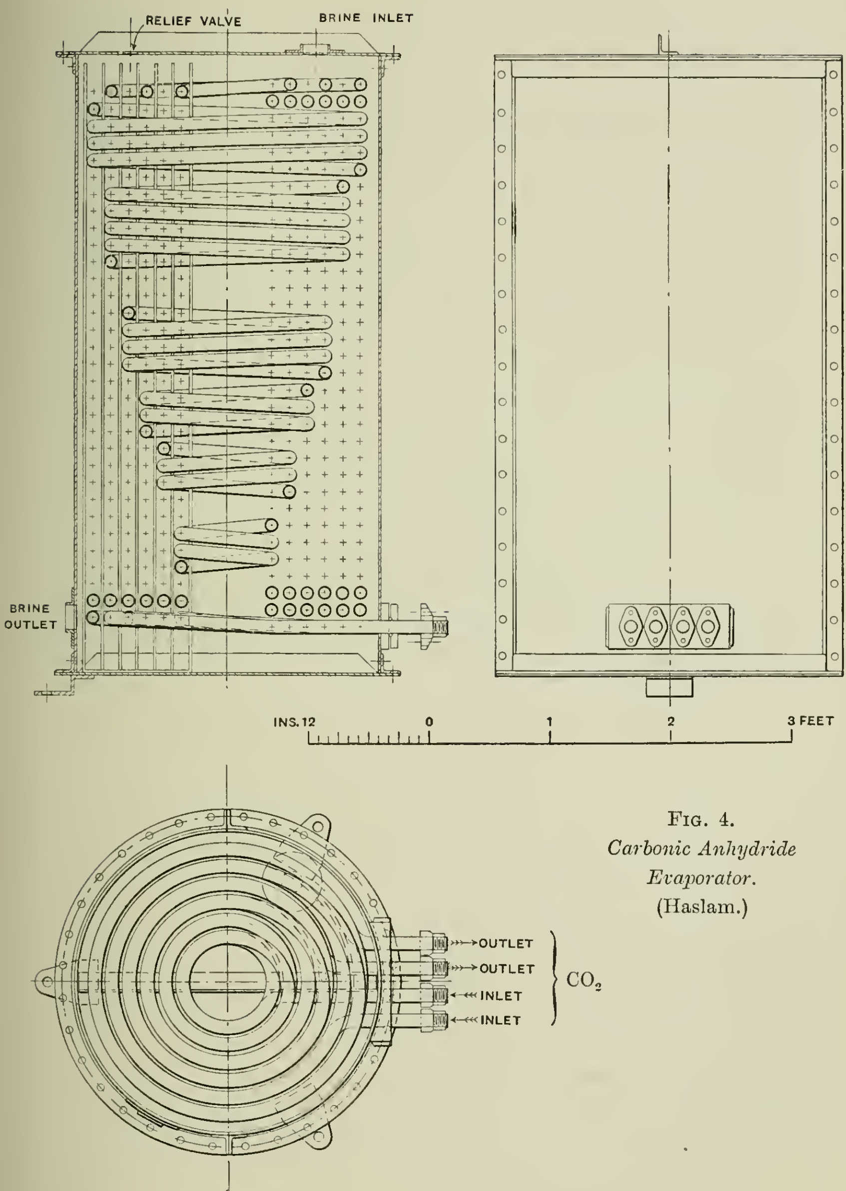



Fig. 4.
*Carbonic Anhydride Evaporator.*
(Haslam.)

between the refrigerant and the brine is not likely to exceed 20° F. Evaporators are often fitted with agitating paddles (*see* Fig. 25, page 989) to keep up an effective "scrubbing" action between the brine and the walls of the pipes. A cover fitted with a relief-valve is shown, as the evaporator, Fig. 4, is intended for marine use, and therefore a closed brine circuit is desirable. The coils are carefully stayed in position by plates and clips. This type of evaporator is also extensively used for land purposes, the cooled brine being afterwards circulated either in pipes placed in the chambers of the cold stores or through ice-tanks for ice-making.

Evaporators take varying forms to suit particular requirements, and some of these forms will be described later in the Paper under "General Arrangements" (page 981).

*Condensers.*—Condensers for vapour-compression machines may be divided into three main classes:—

(1) Submerged;
(2) Atmospheric;
(3) Double-pipe.

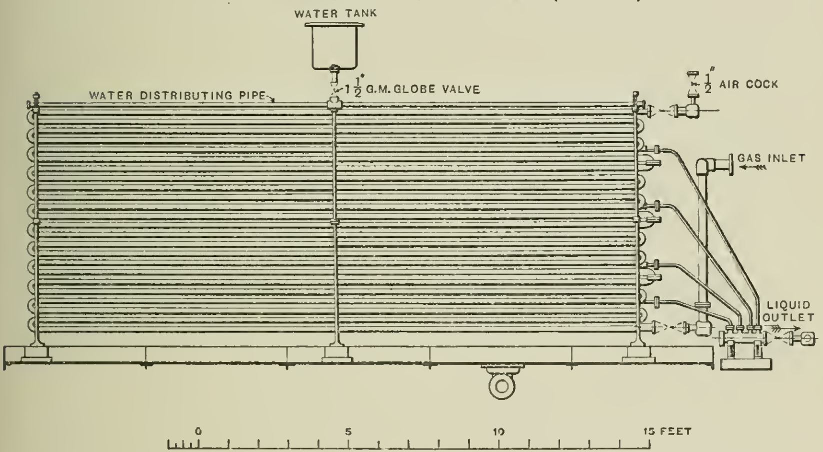
In all three classes the refrigerant is kept inside small tubes or pipes (seldom exceeding $2\frac{1}{2}$ inches diameter), the condensing medium being on the outside. In this respect the condenser resembles the evaporator—indeed, for marine purposes, connections are often arranged so that the two may be interchangeable.

The submerged condenser is either rectangular or circular in plan. In the former case the pipes (up to $1\frac{1}{2}$ inch diameter), in continuous lengths, are bent backwards and forwards within the given length to form grids or elongated coils. This type is commonly used in marine work, the grids being fitted into box-shaped bed-plates, which carry the engine and compressor. Submerged condensers may also be made of 2-inch or $2\frac{1}{2}$-inch pipes, screwed and then soldered into return-bends of $3\frac{1}{2}$-inch or 4-inch centres, the construction of the grids being similar to those in Fig. 5.

The circular form of condenser is similar to the evaporator, Fig. 4, just described. The hot gas enters a manifold at the top and is distributed through concentric coils, the liquid collecting

through a manifold at the bottom. The water enters at the bottom and flows out at the top—the liquid refrigerant being thus cooled to the lowest possible temperature. Theory and practice indicate that the cooling of the liquid refrigerant, after condensation, to the coldest temperature reached in the plant is a distinct advantage—in but few machines, however, are even the most elementary steps taken to secure this advantage. Submerged condensers are only to be recommended when water is plentiful and cheap—as on shipboard.

Fig. 5.—*Contraflow $NH_3$ Condenser.* (Haslam.)




When water is a consideration, considerable economy can be effected by using an atmospheric condenser, that is, condensers where either the air *or* water and air combined perform the necessary cooling. They consist of grids of pipes (1-inch or 1½-inch pipes in continuous lengths or 2-inch to 2½-inch pipes built up by return bends) stacked together in vertical sheets. If air is to be the cooling agent, the sheets must be boxed in and air circulated by a fan or fans—a method adopted in the United States. If water is used—the ordinary British method—it is allowed to trickle down the grids into a pan or tank underneath, the coils being protected from the direct rays of the sun (if necessary) by louvre boards. The water may run direct to waste or be used over and over again.

At Messrs. Ruddin's Central Cold Stores erected in the heart of Liverpool, the author has found that an exceptionally large tank (22 feet $\times$ 11 feet $\times$ 2·5 feet) for the size of the plant has resulted in a most marked economy—the amount of water used in a week not exceeding that generally required per hour—the reasons being the utilization of rain-water and the natural cooling of the water in the tank during the night.

Fig. 5 shows an atmospheric contraflow ammonia condenser. The vapour from the compressor is carried to a tee or manifold at the bottom and there distributed to vertical grids. Water falls on the outside and consequently the warmer water meets the warmer vapour. Liquid will, of course, be formed at varying heights, and small pipes are fitted at different points in the grid to carry off the liquid as soon as formed to the liquid tee or manifold.

Fig. 6 illustrates a special type of condenser that is rapidly finding favour where conditions are not suitable for either atmospheric or submerged condensers. It is known as the "double-pipe" condenser, and in this example $1\frac{1}{4}$-inch water-pipes run through 2-inch ammonia-pipes—the details of construction can be readily followed from the drawing. The hot vapour is brought in at the top and the cold water at the bottom, the whole forming an effective and efficient condenser.

It must be here mentioned that exactly similar constructions are being utilized as evaporators or "brine coolers" and even triple pipes are employed—the refrigerant passing through the annular space so formed. Particular forms of condensers will be mentioned later when describing "General Arrangements."

*Compressors.*—Compressors may be divided into two main classes, namely:—

(1) Double-acting.

(2) Single-acting.

These two classes may be again sub-divided into two sections, namely:—

(i) Horizontal.

(ii) Vertical.

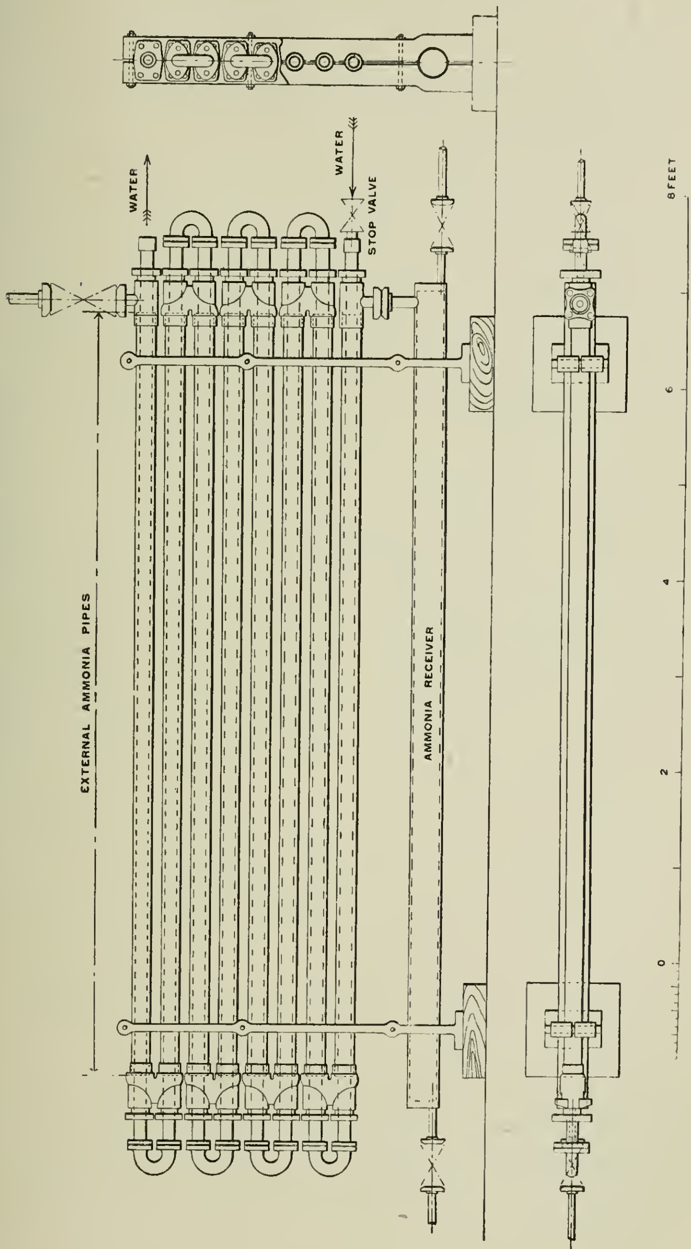



FIG. 6.—*Double-Pipe $NH_3$ Condenser.* (Haslam.)

Broadly speaking, horizontal compressors are double-acting and the vertical, single-acting.

A fairly uniform piston-speed is maintained in all cases, and consequent on the fact that horizontal machines have a longer

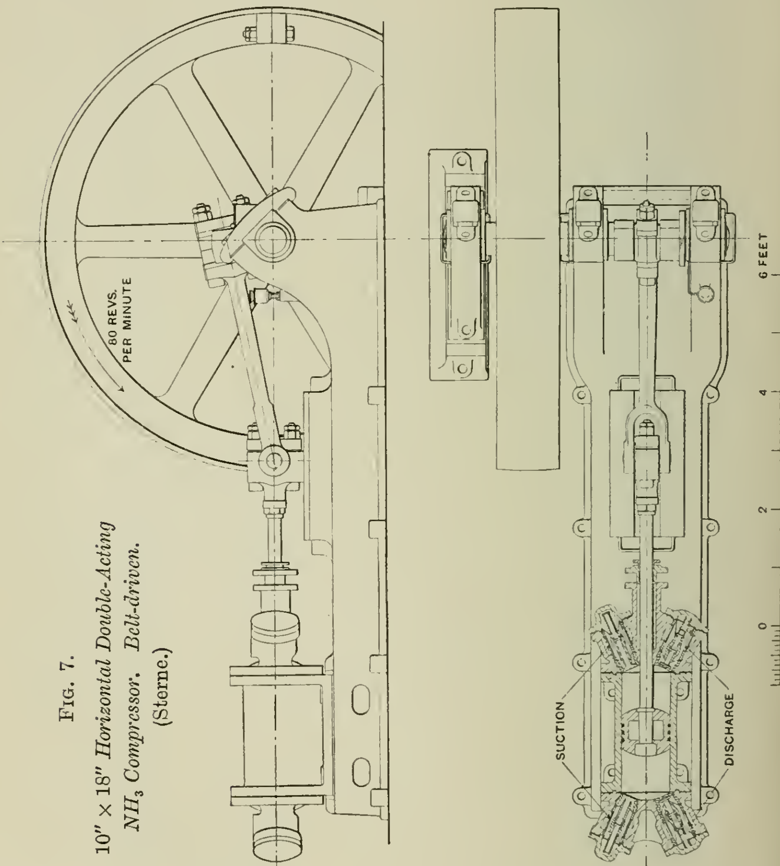



FIG. 7.
*10" × 18" Horizontal Double-Acting $NH_3$ Compressor. Belt-driven.*
(Sterne.)

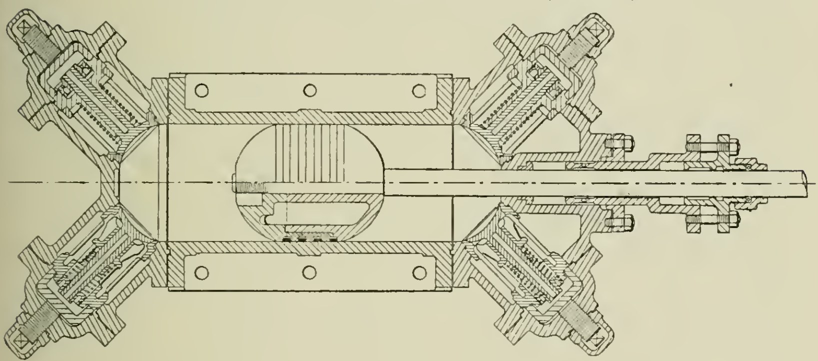
stroke (compared to diameter) than the vertical ones, it follows that the latter type run at a greater number of revolutions, and in certain makes, quite a high speed (revolutions) is maintained.

Apart from purely mechanical considerations, one point in the

design of a compressor stands out as all important, that is, the clearance between the piston and covers at each end of the stroke must be a minimum—the reason being that the compressor should, if possible, draw on the suction-valve immediately the return stroke begins. The vapour that remains trapped in the clearance space is at the superior pressure of the system, and the piston must travel a very definite distance before this vapour can expand below the suction or inferior pressure, and so allow the fresh charge to be admitted.

Apart from making the clearance as small as possible and keeping the bearings and all other parts mechanically sound, nothing further is done as a general rule. An outstanding exception

FIG. 8.—*Section of $NH_3$ Compressor.* (Haslam.)



is the "De la Vergne" system, where vertical compressors are used, and the top of the piston carries a layer of oil sufficiently deep to ensure that the whole of the vapour (ammonia) is forced out at the end of the stroke.

Fig. 7 is a 10-inch diameter by 18-inch stroke horizontal double-acting belt-driven ammonia compressor. The main casting is a simple one, being a plain cylinder provided with feet or lugs for bolting to the bed-plate, no jacket or liner being fitted. The covers carry the suction and discharge valves and the cover at the front also carries the stuffing-box. The valves are arranged so as to be easily accessible, and on the removal of a cover a sleeve can be withdrawn containing the whole of the valve and

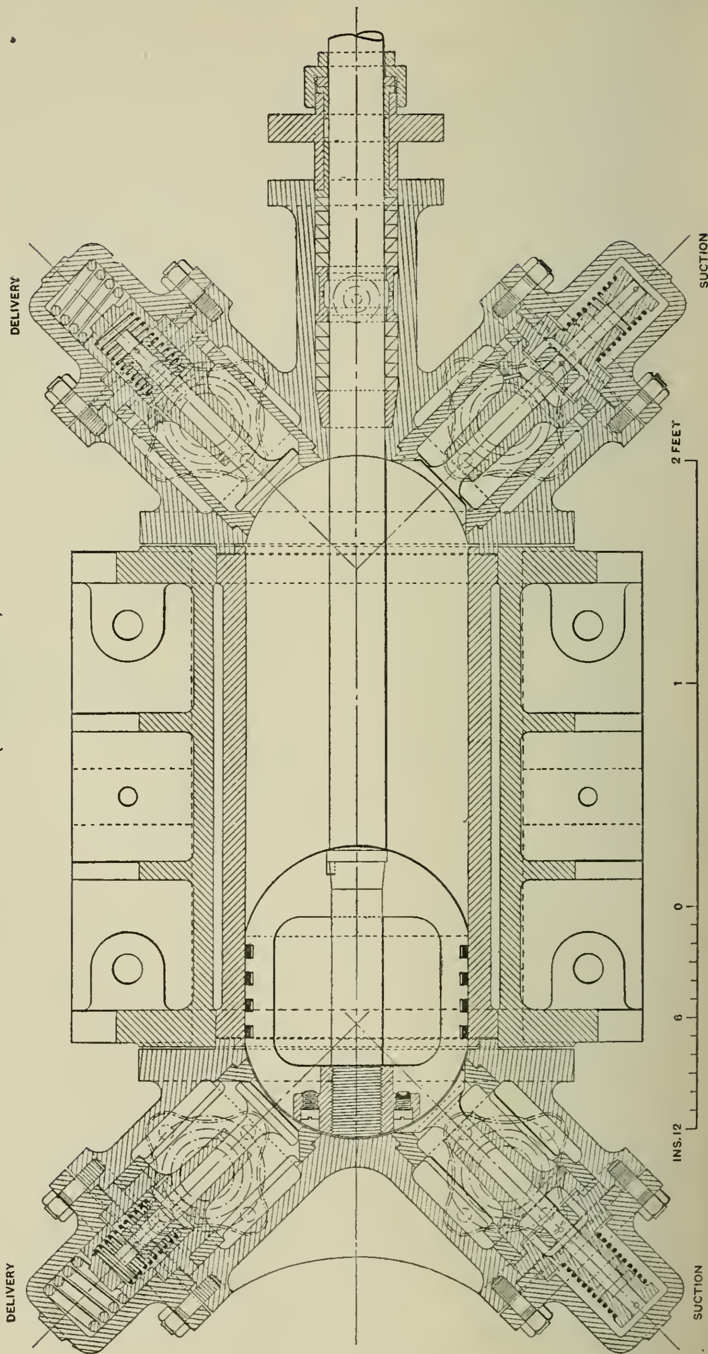



Fig. 9.—*Arrangement of Compressor Cylinders, Piston and Valves for 12″ × 21″ Triplex $NH_3$ Compressor.*
(The L. R. Co.)

its parts. The machine is designed to run at 80 revolutions per minute, and the scale will allow the leading sizes and the floor space occupied to be determined. Photos of similar compressors are shown on Plate 42.

Fig. 8 shows the type adopted by Messrs. Haslam (Derby) for all their ammonia compressors. In general outline it is very similar to Fig. 7, and, taken with Fig. 9, it illustrates the degree of standardization which has been reached in horizontal ammonia compressors. The valves, however, differ in important detail, and, together with the stuffing-box, will amply repay careful examination.

Fig. 9 is a 12-inch diameter by 21-inch stroke horizontal double-acting ammonia compressor. The main difference between this Figure and Figs. 7 and 8 lies in the liner, which is fitted and kept in position by the covers. This arrangement has all the advantages generally claimed for liners, and the claim that so simple a casting can be made with a specially close-grained cast-iron is one of great importance, inasmuch as ammonia readily finds out any slightly extra porosity. The need of reducing the clearance space to a minimum has already been emphasized, and, as the suction valves open inwards, particular care is taken that these valves are well guided so as to prevent any possibility of "sticking." It should be further noted that the weakest part of the suction-valve spindle is at the end in the thread under the nut controlling the spring. If the spindle should break at this point, a nut lower down is so arranged to prevent the valve dropping into the compressor.

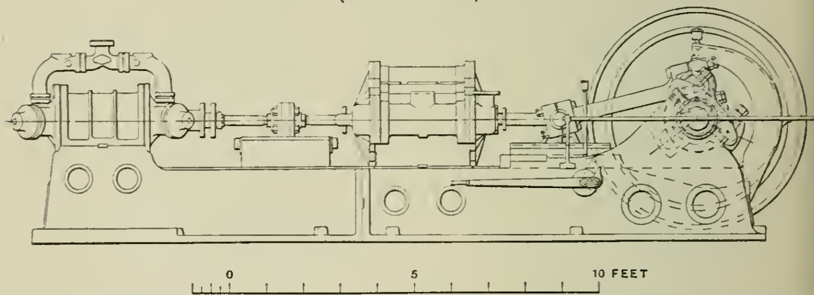
The lift of the discharge-valve is regulated by a stop kept in position by a strong spring. This serves as a buffer and reduces the shock or jar, and further, on occasion, will allow the valve an increased lift should an undue pressure (say from liquid ammonia) be set up at the end of the stroke. The stuffing-box is arranged for metallic packing, and in the main box will be noticed a sleeve designed to leave an annular space between the rod and the box. This space is connected (*see* Fig. 11, page 971) with the suction side of the compressor and fulfils two functions—first, it takes back any high-pressure ammonia that might have so far leaked past, and, secondly,

it provides that the remainder of the box shall only be under suction pressure. This ingenious and simple device is adopted by most makers.

Fig. 10 shows an elevation of Fig. 9—three such compressors being driven direct from the tail-rods of the cylinders of a triple-expansion engine—the whole forming a powerful and compact installation for shipboard. A photo of the machine is given on Plate 43.

Extra precautions have to be taken in refrigeration to ensure (particularly in marine work), as far as possible, immunity from complete breakdown, and in Fig. 10 the possible combinations in case of necessity are too numerous to mention, but it must be stated

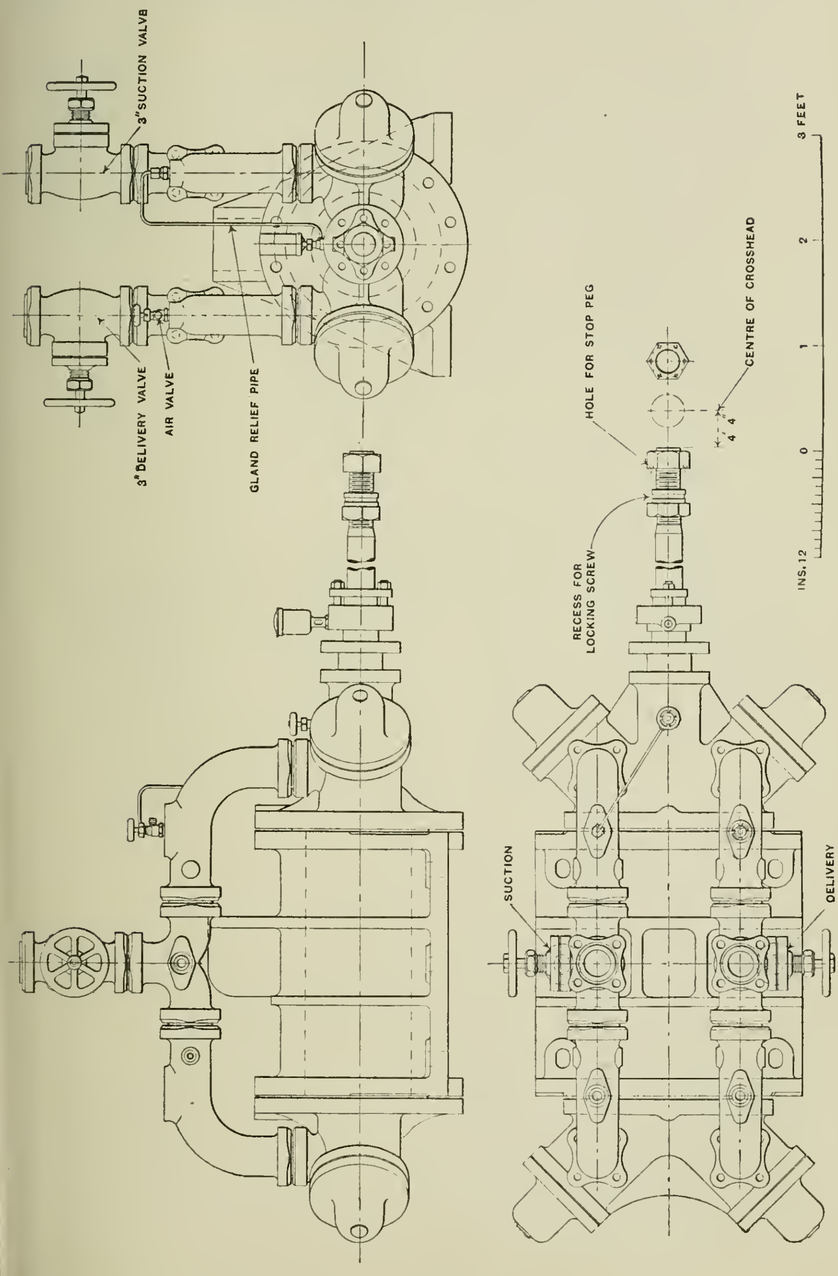
Fig. 10.—12″ × 21″ *Triplex $NH_3$ Compressor.* (*See also* Plate 43.)
(The L. R. Co.)




that any two cylinders of the engine can be worked together compound, and also any single cylinder by itself; further, as long as the crankshaft and rods hold out, any one cylinder will drive any one compressor. Small faces are provided for gauge and similar connections indicated in Fig. 11.

Fig. 11 gives the details of connections similar to those fitted to the compressors given in Figs. 9 and 10, although the actual dimensions are for a longer stroke, namely, 12 inches by 24 inches. The arrangements are so clear in the drawing as to call for little or no comment, but the stuffing-box relief connections should be noted and also the lubricator to the rod, which is so arranged as to keep a space in an outer gland (*see* Fig. 9) flooded with a special oil, a further small gland

FIG. 11.—*Arrangement of Compressor Cylinder.* 12″ × 24″ $NH_3$. (The L. R. Co.)



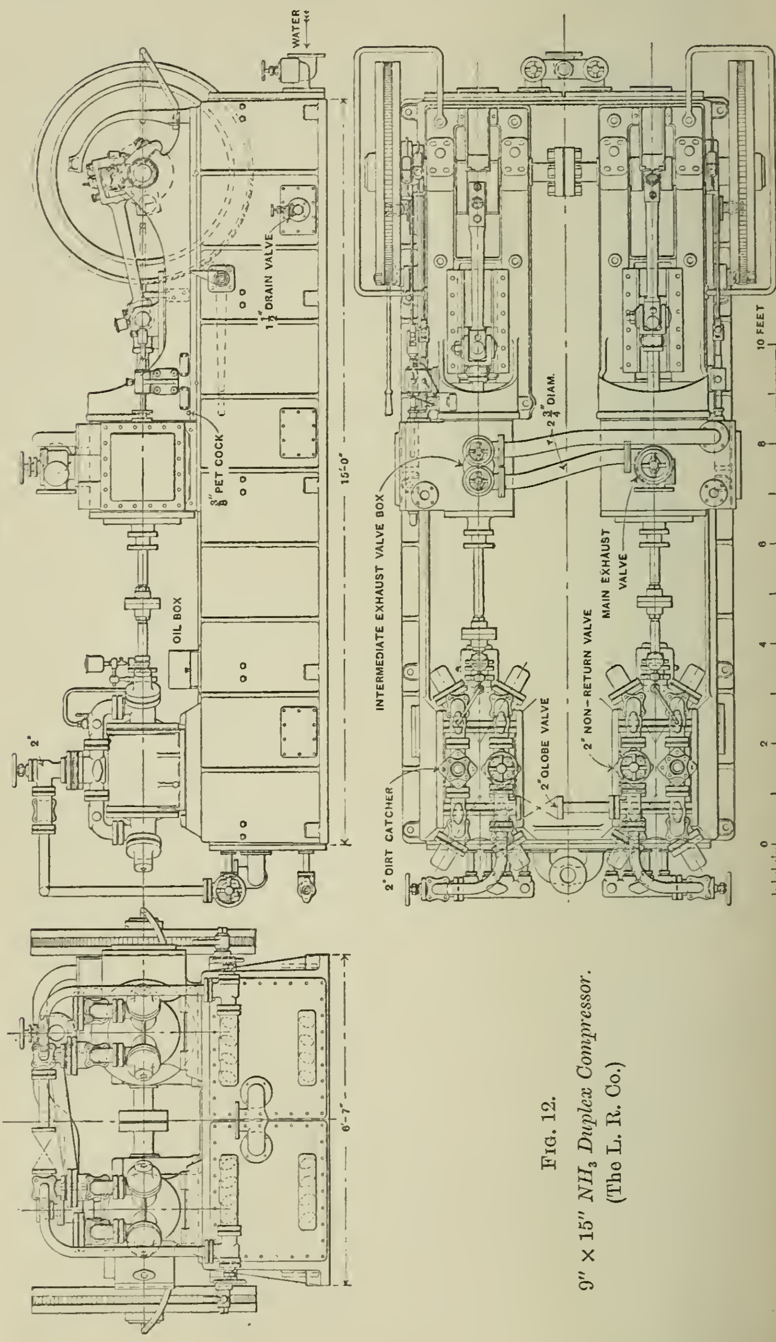



FIG. 12.

9″ × 15″ $NH_3$ *Duplex Compressor.*

(The L. R. Co.)

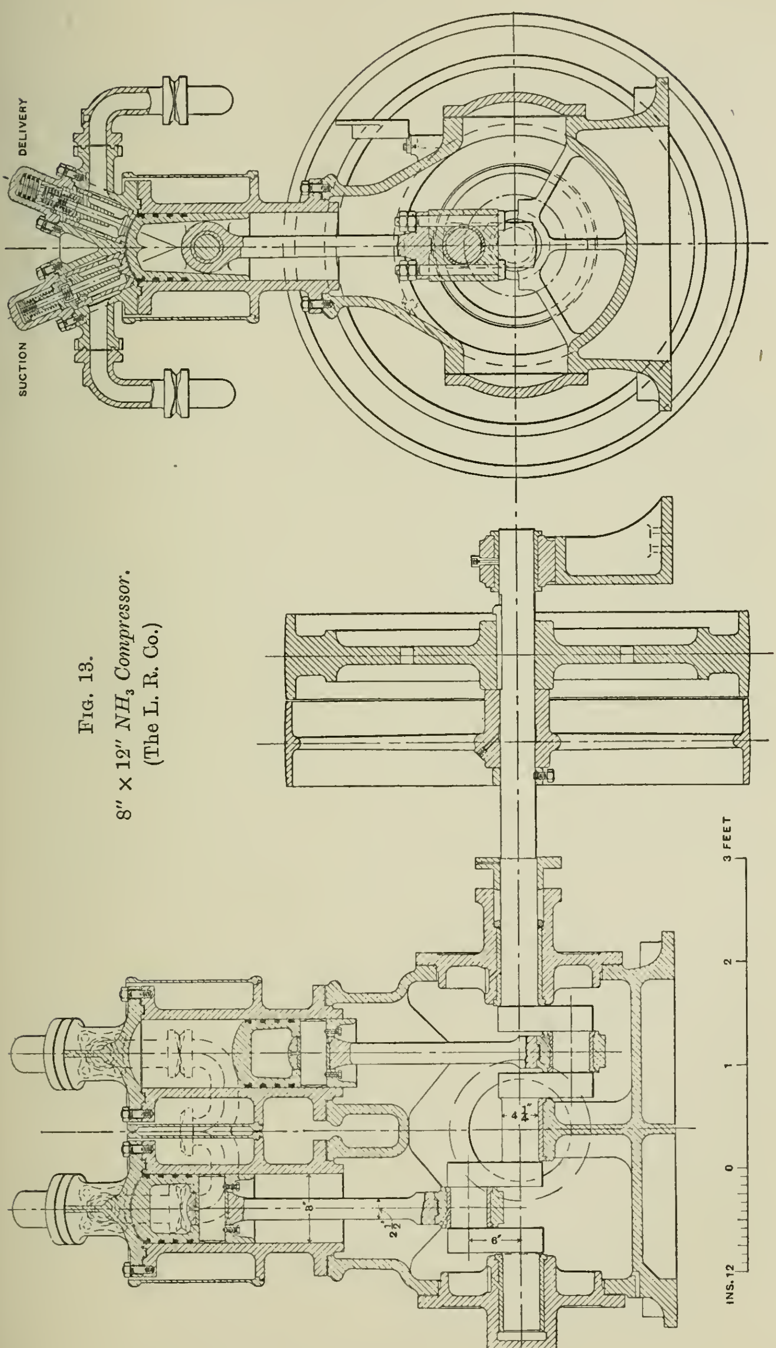



Fig. 13.
8" × 12" $NH_3$ *Compressor.*
(The L. R. Co.)

keeping this oil in place. It is sometimes an advantage to so arrange the suction connections, that each end of the compressor (*see also* Fig. 25, page 989) can draw from separate evaporators—it being remembered that temperatures are regulated by suction pressures—and different suction pressures could be carried in the present case by having a separate connection with valve for each suction end. A cross connection with valve would also enable the full compressor to draw from any one evaporator or from both under the same suction pressure.

Fig. 12 shows an arrangement of two 9-inch diameter by 15-inch stroke ammonia compressors by the same firm as the three previous figures. The main difference (apart from the number of compressors) between this arrangement and that shown in Fig. 10, and the reason for its introduction, is that two separate and distinct ammonia condensers—one for each compressor—are placed in the box bed-plate, this being in accordance with ordinary marine practice, a cross connection allowing any one compressor to work with any one condenser. In ordinary working, the two condensers are often worked as one whole. The ammonia and water connections can be readily followed and call for no description. The liquid connection to the evaporator is shown at the bottom left hand of the elevation. A photo of a similar arrangement is given on Plate 43.

Fig. 13 is an 8-inch by 12-inch vertical ammonia compressor. On shipboard, while admitting space is always restricted, yet the greatest restriction lies in the height between decks, and hence, as seen in Figs. 10 and 11, the more powerful marine plants are fitted with horizontal compressors. In certain business premises on land, more particularly in the heart of a city, it is the floor space that is the most valuable, the height being in no case so restricted as on shipboard. In such locations, particularly with a belt drive, a vertical compressor is the better type to employ. In main design the Figure is very similar to the horizontal compressors previously described.

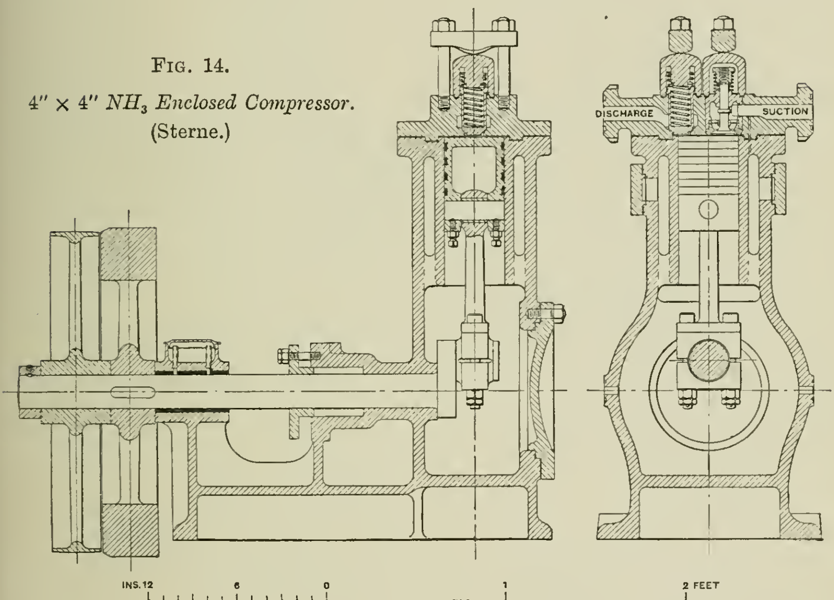
The whole of the working parts are enclosed, and a portion of the crank-chamber to just above the shaft is filled with special (low-point freezing) oil, the remainder of the space being connected to the suction circuit, the usual pressure of which may be taken as

15 lb. per square inch above the atmosphere. The shaft is provided with a stuffing-box, the function of which is to prevent the oil from leaking out, the oil effectually sealing this possible source of leakage from the ammonia gas. The whole arrangement as shown is equivalent to one 8-inch by 12-inch double-acting compressor, but, whereas the double-acting compressor would not likely run above 80 revolutions per minute, this compressor is designed to run

Fig. 14.

4″ × 4″ $NH_3$ *Enclosed Compressor.*

(Sterne.)




at 120 revolutions per minute. A photo of the machine is shown on Plate 42.

Fig. 14 is a vertical ammonia compressor. The design, although distinctive, follows much the same lines as those previously described, and oil is used as in Fig. 13. The main point of note is the short stroke, which is the same as the diameter, namely, 4 inches. It is frequently found that high-speed shafting or an electric motor happens to be the most convenient source of power, and in such cases a quick-revolution compressor is a distinct

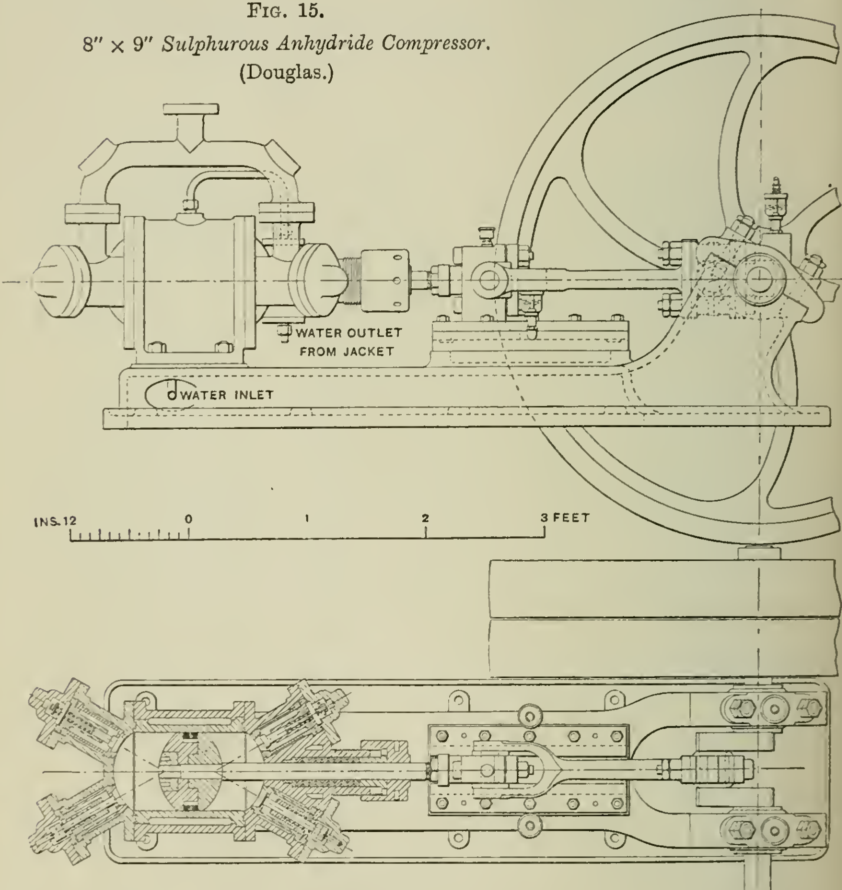
advantage. In the present case the compressor is designed to run at 200 revolutions per minute as a maximum.

Fig. 15 is an $SO_2$ horizontal double-acting belt-driven compressor, 8 inches diameter by 9 inches stroke. The same main features, as

Fig. 15.

8″ × 9″ *Sulphurous Anhydride Compressor.*

(Douglas.)




previously outlined in ammonia compressors, are to be observed. The construction is simple, the compressor-wall being a liner with a water-circulating space between it and the main body casting. The water is first circulated round the liner and then by means of a

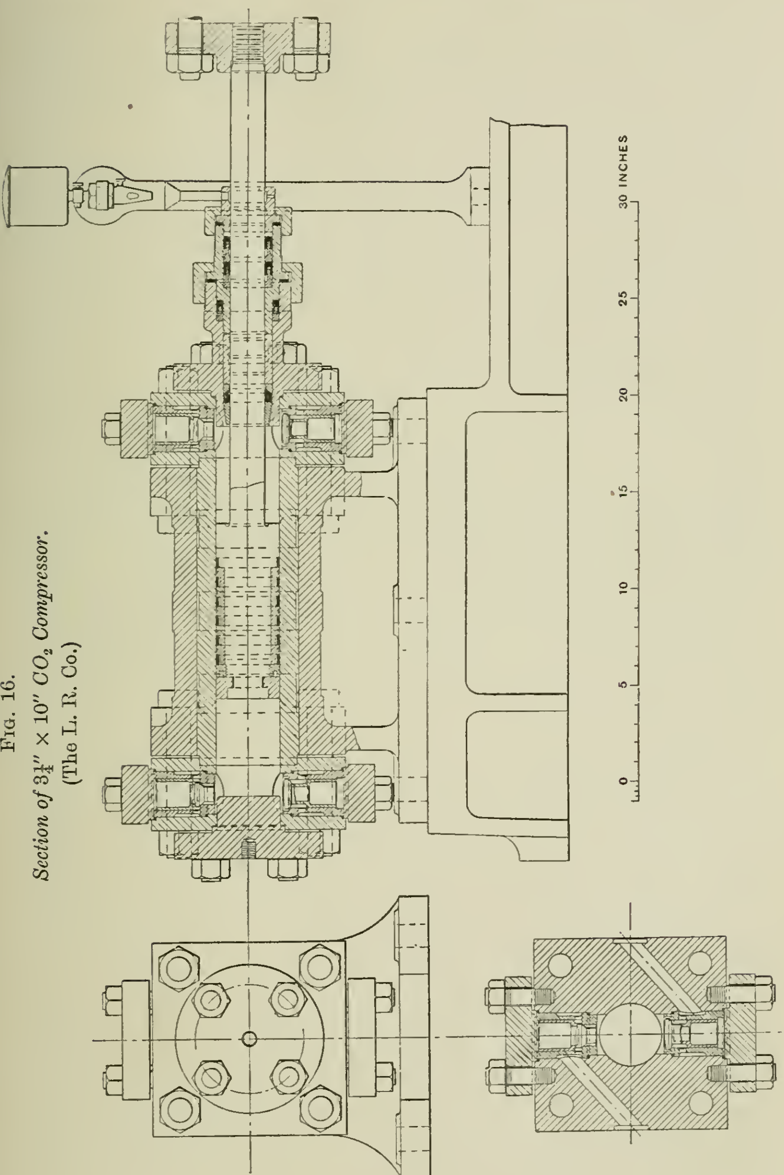



Fig. 16.
*Section of $3\frac{1}{4}'' \times 10''$ $CO_2$ Compressor.*
(The L. R. Co.)

small pipe, clearly indicated, is taken to the front cover and circulated round the compressor end of the stuffing-box. This box is of the long single type, and is all that is required for the pressures to be met. The whole design is simple and makes a very effective machine.

Fig. 16 is a $3\frac{1}{4}$-inch by 10-inch $CO_2$ compressor. It consists of special cast-iron liner fitting for practically its whole length into a cast-steel body; the end-covers which carry the valves and keep the liner in position are of mild steel, the stuffing-box being bolted on as a separate piece on the front end. The very high condensing pressure of $CO_2$ (average about 900 to 1,000 lb. per square inch) calls for special care in the stuffing-box, and in the present case it is arranged to supply the centre of the box with oil at a pressure slightly higher than the condensing pressure. Cup leathers (woodite rings) prevent this oil from passing outwards, and it tends to pass into the compressor, the whole design effectively preventing the escape of $CO_2$. A special oil-trap (*see* Fig. 18, page 980) is arranged to catch this oil on the discharge pipe.

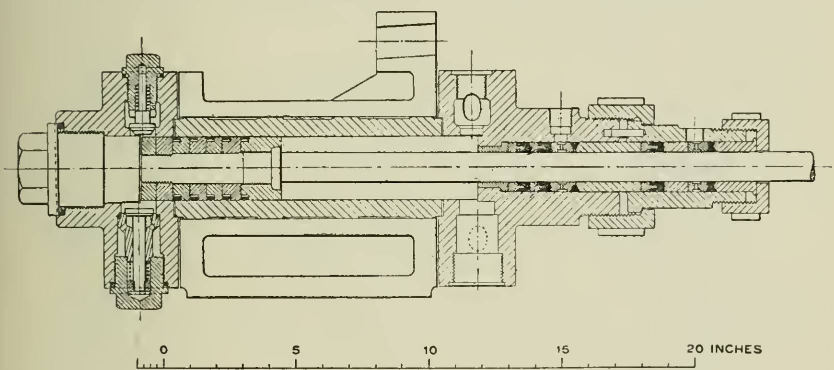
The great difference between the evaporator (suction) and condenser (discharge or delivery) pressures (say 600 to 700 lb. square inch) means that the greatest care must be exercised in the finish of the bore of the compressor and also in the design and manufacture of the piston. In the present example the piston and piston-rod are turned from a single bar of special steel, the six packing-rings being threaded on in five rings, kept in position by a ring at the end made in halves, the two parts simply slipping into position. The clearance spaces, on account of the position of the valves, cannot relatively be so finely adjusted as in ammonia compressors, but the greater ratio that always exists in $CO_2$ machines between the diameter and stroke compensates for the clearance loss.

Fig. 17 is a $CO_2$ compressor, $2\frac{3}{8}$ inches by $7\frac{1}{2}$ inches. In the main this compressor is similar to Fig. 16, and again serves to show how the result of practical experience tends to bring about a standardization of design in machines performing exactly the same class of work. The details of both the stuffing-box and valves differ slightly from Fig. 16, but they are so clearly shown that the construction can be readily followed.

Fig. 18 (page 980) is a general arrangement of what is known as a $4\frac{1}{2}$-inch by 15-inch $CO_2$ "duplex machine" for marine purposes, that is, the double-acting compressors, each of the sizes given, are arranged to work as one machine with a common condenser in the box bed-plate, but each compressor can, if desired, work alone with its own separate condenser, while in the case of a breakdown in one or more sections of the machine there are several possible cross combinations.

Although of larger size the views given of the compressors may be taken to supplement those given in Fig. 16, and, together with

FIG. 17.—*Section of* $2\frac{3}{8}'' \times 7\frac{1}{2}''$ *$CO_2$ Compressor.* (Haslam.)




the connections, are so clear that detailed description is unnecessary. The pressure lubricator, however, deserves special mention. It consists essentially of a little plunger-pump. The vapour at the superior pressure from the discharge-pipe is allowed to act on the side of the plunger remote from the rod. On the other side, in the annular space between the rod and the barrel, there is a special oil, which, on account of the difference in area, is forced along a small copper pipe to the stuffing-box, at a pressure slightly higher than the discharge pressure of the machine, the result being, as previously explained, the $CO_2$ is prevented from leaking outwards, the oil rather leaking in. The condensers, as mentioned, are formed by the box bed-plate and solid-drawn copper pipes made up

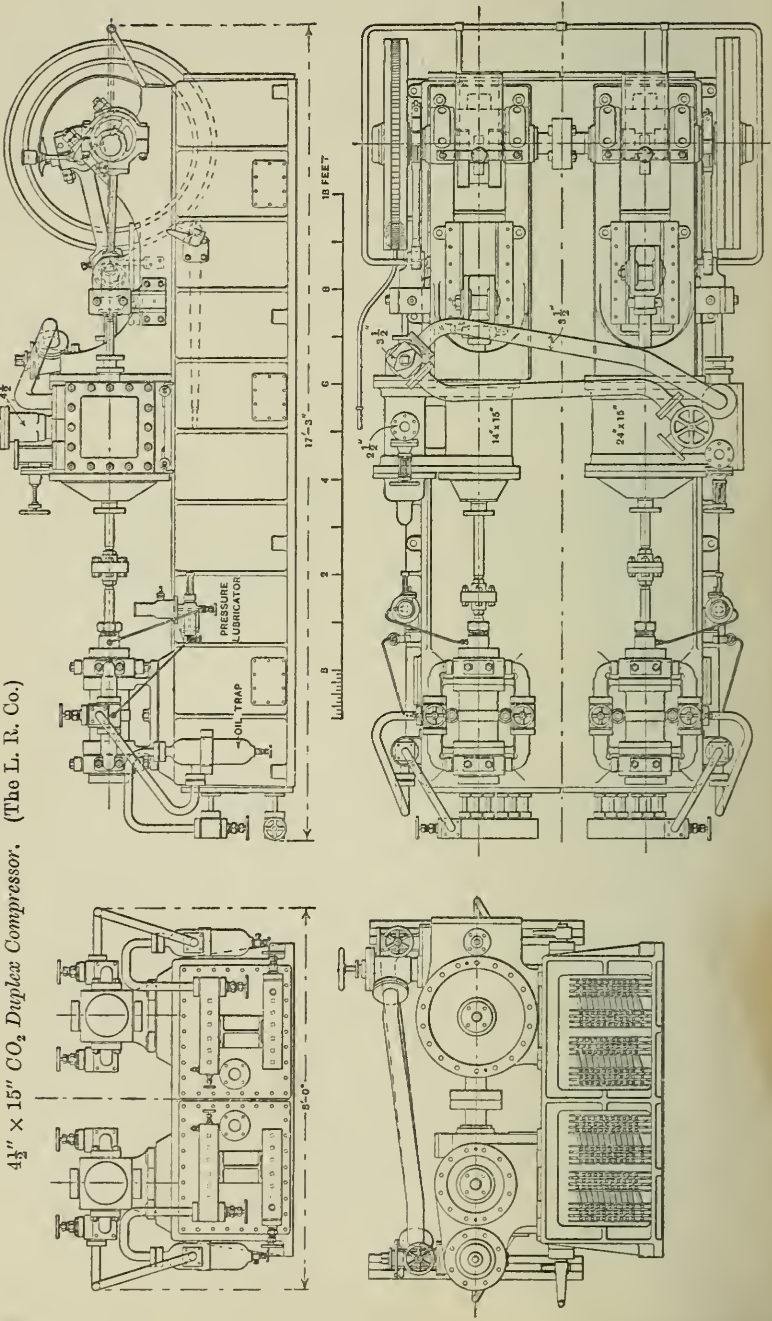


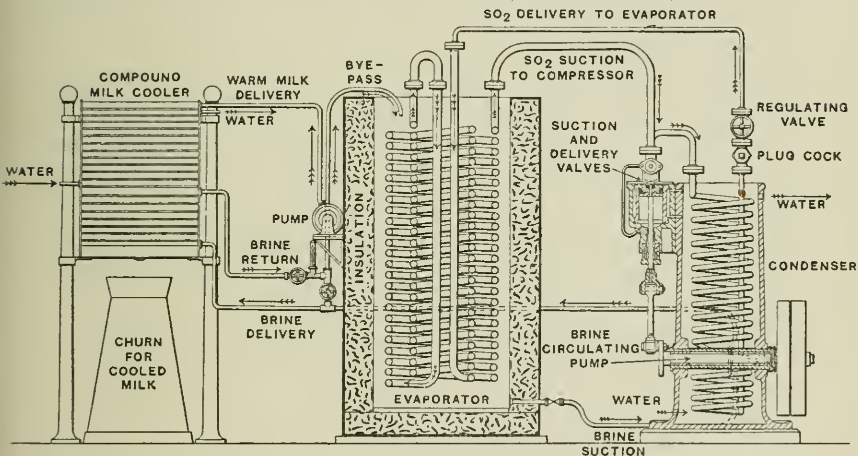

FIG. 18.

$4\frac{1}{2}'' \times 15''$ *$CO_2$ Duplex Compressor.* (The L. R. Co.)

into oval coils placed in sets, one inside the other, to economize space, six separate circuits being arranged in each condenser all connecting to headers or manifolds at the top and bottom, the whole making a very compact machine, a photo of which is given on Plate 43.

Vertical $CO_2$ compressors are also extensively used, but they do not present any distinctive features from those already described—excepting, perhaps, that they are frequently cast in bronze. Photos of vertical $CO_2$ compressors are given on Plate 44. Some makers

Fig. 19.—*Sectional Diagram of a $SO_2$ Refrigerating Machine for Milk Cooling by means of a Compound Cooler.* (Douglas-Conroy.)




cut both horizontal and vertical $CO_2$ compressors from a solid block of mild steel; an excellent example is shown in the bottom photo, Plate 44.

*General Arrangements.*—Fig. 2 (page 957), as already described, indicates diagrammatically the cycle of operations in a compression machine; it is now proposed to examine, as typical examples only, a few actual general arrangements. It should be noted that although methods, uses, and details vary greatly, there is no departure whatever from the cycle of the refrigerant.

Fig. 19 and Plate 45 show a milk-cooler. A single-acting, water-jacketed, vertical $SO_2$ compressor is mounted on the condenser which encloses a single coil of pipe. The vapour from the compressor enters the top of this coil, the resulting liquid passing away from the bottom by an internal vertical pipe to the regulating valve. The condensing water enters at the bottom and leaves at the top. After passing the regulating valve the $SO_2$ is taken direct to the bottom of an internal coil in the evaporator; the top of this coil is connected with the bottom of a larger coil, and after passing through this to the top the expanded vapour is led by a suction pipe back to the compressor. Brine is circulated round the coils and supplies the heat necessary for the complete evaporation of the $SO_2$, the brine being thereby cooled.

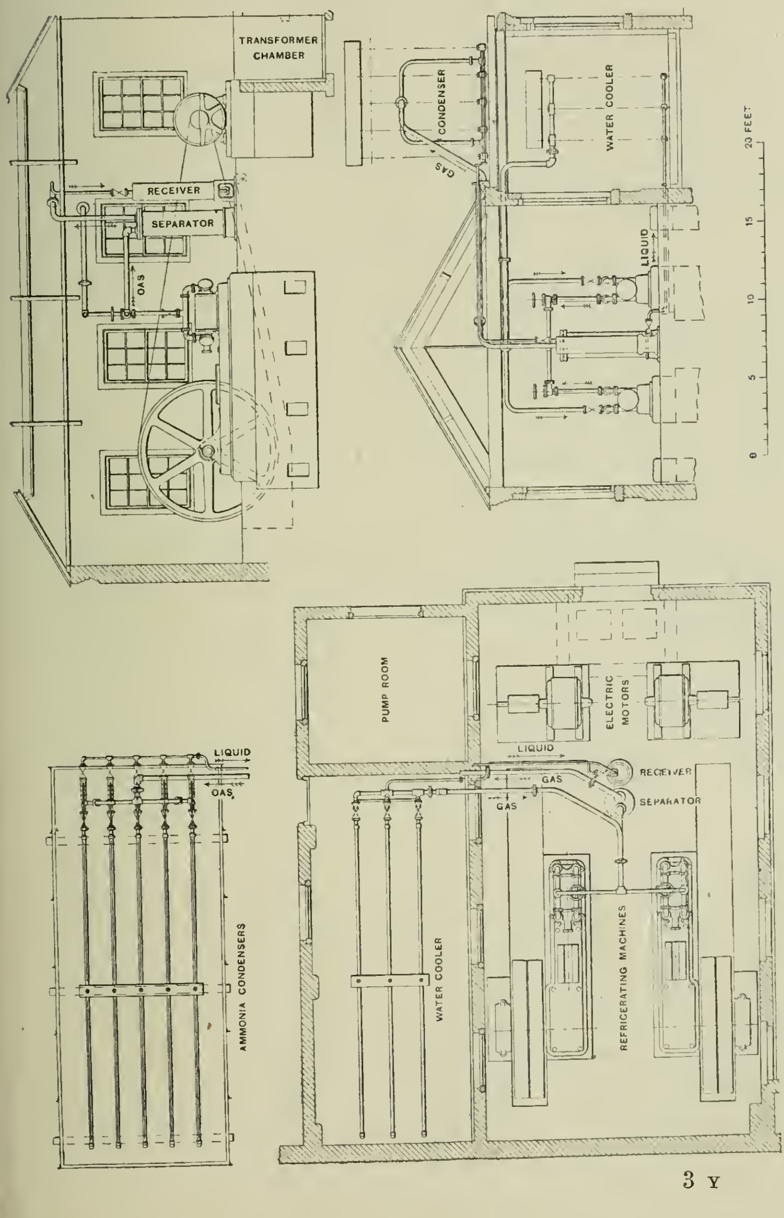
Through the top portion of the actual milk-cooler water is circulated, and through the lower portion the cold brine from the evaporator is passed—the water doing the preliminary cooling and the brine the final cooling of the milk. The milk is thus the *heat inlet* to the machine, the complete heat circuit being:—milk to brine in the cooler, brine to $SO_2$ in the evaporator, $SO_2$ to the condensing water. The condensing water is therefore the *heat outlet*, and the total heat carried away would be that taken from the milk plus the heat equivalent of work expended in the compressor plus the heat leakage into the machine and its connections. (For the approximate amounts of these three quantities of heat in various sized machines, *see* Appendix V, page 1027.)

The chief mechanical details can be easily followed from the Figure, but it should be noted that, by an arrangement of two valves and a by-pass overflow pipe, the amount of cold brine flowing through the cooler can be so regulated that the milk runs off the cooler at the desired temperature.

The insulation around the evaporator is to prevent an undue influx of external heat to the brine. A photo of a complete $SO_2$ machine for land use is shown on Plate 45.

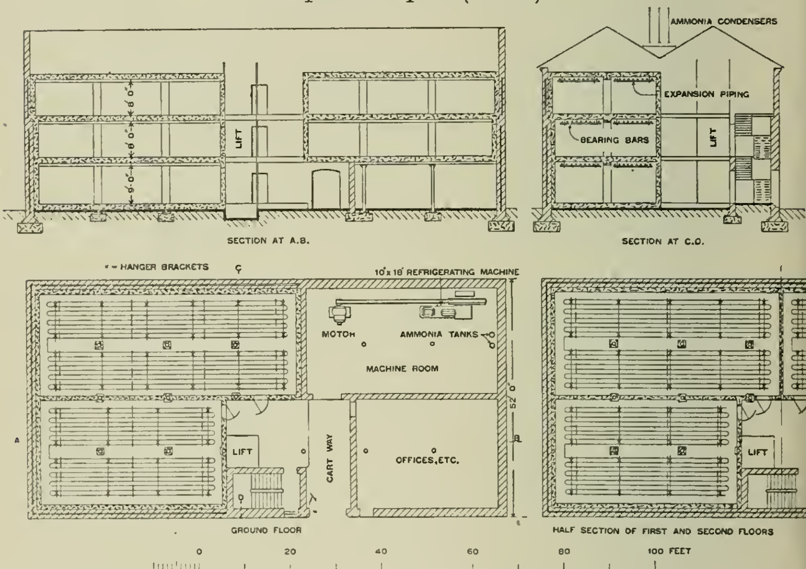
Fig. 20 is a water-cooling plant for a brewery. Here the ammonia compressor is duplicated as being the part requiring most attention—repairs and re-adjustments—and to avoid, as far

Fig. 20.—*Water-Cooling Plant in Brewery. Arrangement of Piping in Refrigerating Engine Room.* (Haslam.)



as possible, complete stoppage by breakdown. The evaporator consists of a series of horizontal pipes formed into vertical grids, down which, on the outside, the water to be cooled trickles. The ammonia vaporizes inside these pipes by virtue of the heat supplied by the water, the resultant vapour in the pipes being drawn off by the compressor and discharged into the condenser, which is of the atmospheric type; the resultant liquid

Fig. 21.—*General Arrangement of Cold Stores. Cooled by Direct Ammonia Expansion Pipes.* (Sterne.)




gravitates back to a liquid receiver. The drawing is very complete, all the connections being shown, and particular attention is again drawn to the separator placed between the compressor and condenser for catching oil passed over with the ammonia vapour. The capacity of the machine (that is, with one compressor working) is 4,400 gallons of water per hour cooled from 60° to 50° F. or half that quantity cooled from 70° to 50° F.

Fig. 21 shows a general arrangement of a cold storage cooled by what is known as the "direct expansion" method. The evaporator

in this case consists of grids of pipes placed directly in the chambers to be cooled, the air supplying the heat (taken from the goods stored) necessary for vaporization. The compressor (shown in detail in Fig. 7, page 966) draws the ammonia vapour back from these grids and discharges it into an atmospheric condenser placed on the roof. The liquid ammonia gravitates into tanks placed in the engine-room, and by means of suitable tee-pieces and a number of regulating valves the liquid is distributed to the direct-expansion pipes in the separate rooms for re-evaporation. The whole forms a most efficient and economical system.

Figs. 22 and 23 (pages 986–7) show a general arrangement of the machinery for an ammonia plant on shipboard. The compressors are directly driven from a steam-engine, the whole standing on a box bed-plate inside of which are placed the grids of pipes forming the condenser (*see* photo, Plate 44). Two distinct types of evaporator are provided in this installation: (1) Circular coils placed in vertical shells for cooling the brine which is circulated through grids of pipes placed in those holds where *chilled* meat or produce is carried; these evaporators are termed "brine coolers" in the drawing. (2) Longitudinal grids of ammonia pipes in which the liquid vaporizes by the heat brought to the grids by the air which is circulated around them by a fan—the air being drawn up from those holds when *frozen* meat or produce is carried and after being cooled is sent down once more, suitable air-suction and delivery ducts being fitted. These evaporators are termed "air-cooling batteries" in the drawing. The evaporators are placed in insulated spaces and even the cold ends of the brine pumps are also encased.

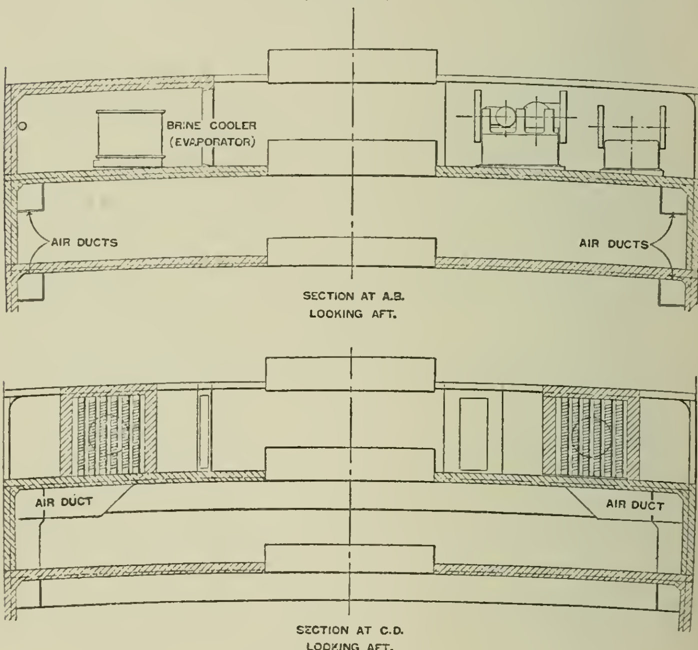
It should be noted that while the air-blast system has much to recommend it, it is impossible by its means to regulate the temperatures carried by this system to the very fine point (one or two degrees variation only) of regulation required for *chilled* produce. Hence the two systems shown in the Figure.

Fig. 24 (page 988) is a general arrangement of a $CO_2$ plant. The compressors are horizontal and directly driven (as in the previous example) by steam-engines. The great value of this

Figure lies in the fact that it shows all the arrangements that have to be made to allow the withdrawal, inspection, and cleaning of all the coils in the machine. These are all so clearly shown as to require no further explanation.

Fig. 25 (page 989) shows the arrangement of an engine-room in Cold Stores at Wigan. An ammonia compressor 11 inches

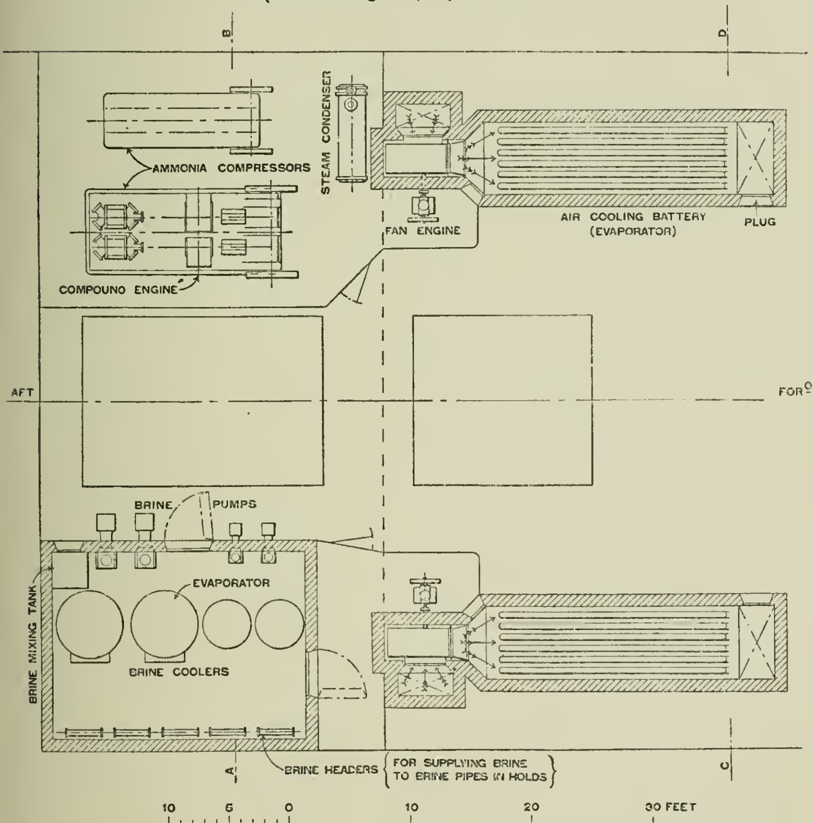
FIG. 22.—*Arrangement of Refrigerating Machinery on Shipboard.* (Haslam.)




diameter by 20 inches stroke is arranged to do all the work required at 60 revolutions per minute, namely, make 12 tons of clear hard ice per day and keep the Cold Stores at the required temperature. A smaller compressor is fitted as a stand-by, the connections (*see* Fig. 25) being so arranged that in case of a breakdown the ice-making is stopped and the Stores kept in condition. Further, the small compressor may be used for winter work on the Stores supposing the ice stock is sufficient to meet the

small demand. Both compressors are separately driven by specially designed electric motors taking current from the Corporation mains; the larger motor is rated for 90 amperes at 460 volts and 300 revolutions per minute, the normal load being 70 to 75 amperes.

FIG. 23.—*Arrangement of Refrigerating Machinery on Shipboard.* (Plan of Fig. 22.) (Haslam.)

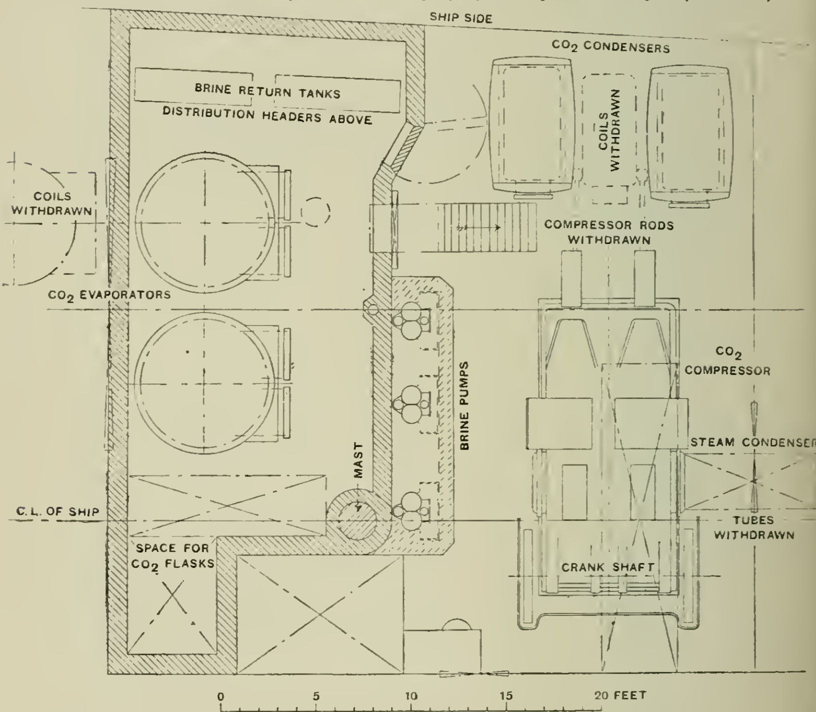



A by-pass connection on each of the compressors enables an easy start to be made by relieving the compression at the end of each stroke for the first minute or so, thus allowing the whole machine to get "under weigh" before the suction valves are opened, thus dispensing with the use of loose pulleys.

This method of starting was first adopted, as far as the author is aware, at Messrs. Ruddin's Central Cold Stores, Liverpool, in 1897, where gas-engines are used, and both at Wigan and Liverpool the result has been most satisfactory and is now commonly adopted.

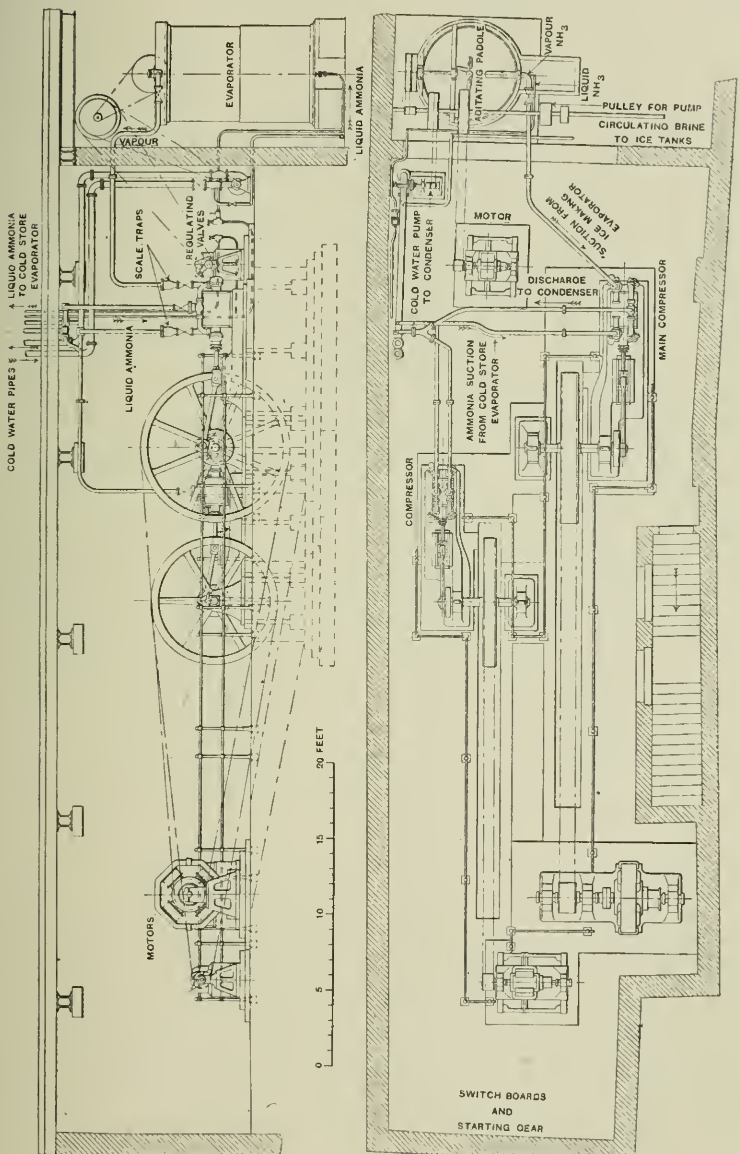
Fig. 25 also shows an evaporator of the circular type with

FIG. 24.—*Marine Arrangement of $CO_2$ Refrigerating Machinery.* (Haslam.)




concentric coils. A central agitator, formed by two radial vanes or paddles secured to a vertical shaft and extending practically the whole depth of the evaporator and driven from the counter-shaft by a belt and bevel gearing, is fitted, to keep up a good "scrubbing" action between the brine and the walls of the ammonia coils; the importance of such action, where the difference of temperature between the refrigerant and the brine is small (seldom exceeding

Fig. 25.—*Arrangement of $NH_3$ Refrigerating Machinery to make 10 tons of Ice per Day at Wigan.*



20° F.), has already been pointed out. This evaporator cools the brine used for making ice on the cell system. The ammonia suction-pipe from this evaporator leads back to *one end* of the main compressor.

The Stores are cooled by blowing air over a stack of brine-cooled pipes which are placed directly over a rectangular shaped tank containing brine, in which the ammonia coils forming the evaporator are placed. The air is circulated by means of a fan electrically driven; specially constructed ducts deliver, distribute, and return the air to the cooler. The ammonia suction from the Cold Stores evaporator (which is placed on the first floor) is taken back to the *other end* of the main compressor, a connection also being made to both ends of the stand-by compressor. The whole arrangement of ammonia connections should be carefully noted, as it affords a typical example of good practice.

The condenser is of the atmospheric type where the ammonia enters the top of the vertical grid-like coils, a very large tank and centrifugal pump (*see* Fig. 25) ensuring a good and constant circulation of water over the pipes. The tank is placed over out-houses in a position exposed to the sun; the coils are in consequence shielded from the direct rays by louvre boards. The connections from the compressors to the condenser and the liquid return to the two regulating valves—one for each of the evaporators—are clearly shown in the Figure. The contractors for the refrigerating plant were The Liverpool Refrigeration Co., and for the electrical power and light installation Messrs. Drake and Gorman, Manchester and London.

Fig. 26 is a sectional profile of one of Messrs. H. and W. Nelson's latest meat-carrying steamers (S.S. "Highland Laddie," and similar boats). The main machinery and boilers are amidships, dividing the insulated (insulation indicated by thick lines) and refrigerated cargo space of about 350,000 cubic feet into two sections—one forward and the other aft. The refrigerating engine-room is aft, and opens out of the main engine-room. The holds are cooled by means of 2-inch diameter galvanized-iron brine-pipes fitted in each hold and between decks—the quantity

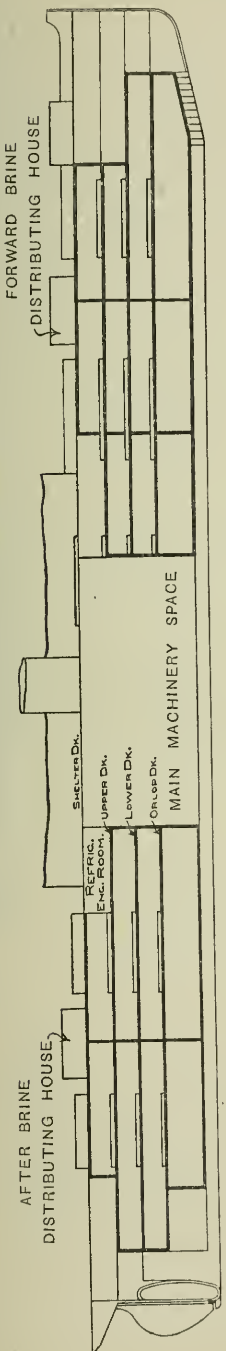



FIG. 26.—*Diagram showing Meat-carrying Steamer.*

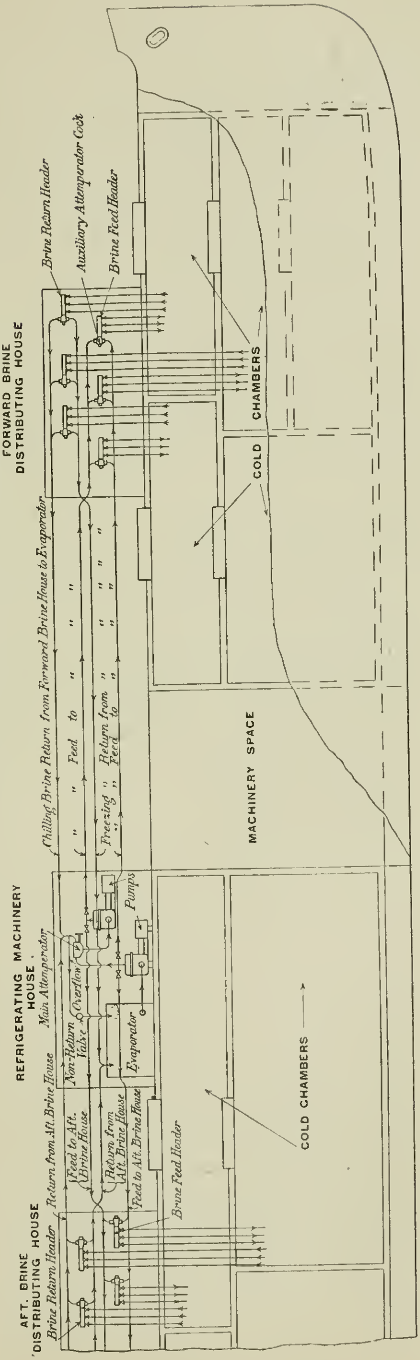
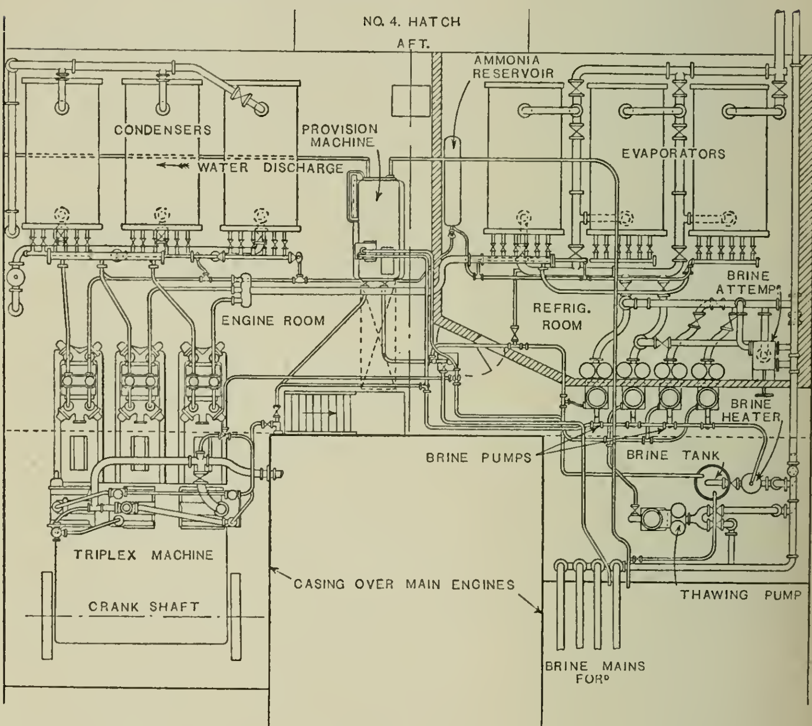



FIG. 27.—*Diagram Arrangement of Attemperated Brine System on Steamer.*

and temperature of the brine for each circuit being regulated from a distributing house. In this installation there are two such houses—one forward and the other aft. These houses are supplied with two distinct feeds of brine—one being at about 5° F., technically known as zero or freezing brine, the other at about

Fig. 28.—*Arrangement of Refrigerating Engine Rooms.* (Nelson Liners.)




35° to 38° F., known as chilling or attemperated brine. There is still a third circuit known as "thawing-off" brine, which is used *after* all the refrigerated cargo has been taken out, the purposes being, first, to thaw off all the ice and snow collected on the brine-pipes in the refrigerated spaces, and, secondly, to dry out thoroughly the holds ready for the outward cargo.

Fig. 28 is a plan of the arrangement of the refrigerating-machine room. Three horizontal double-acting ammonia compressors are fitted, a section of one of which is given in Fig. 9 (page 968), each compressor being driven direct off the tail-rods of a triple-expansion engine; the actual arrangement is shown and has been described under Fig. 10 (page 970). The compressors discharge into three condensers, all the connections of which are so arranged that each condenser can be worked, if necessary, quite independently of the other two and with any one compressor. Sea-water circulated from a pump in the main engine-room is used for condensing purposes. The resultant liquid ammonia is taken to a liquid receiver, placed in the insulated chamber (a practice theoretically and practically sound) containing the three evaporators. These are exact duplicates of the condensers, and consist of rectangular shaped mild-steel tanks, each containing six separate circuits of lap-welded wrought-iron coils, connected top and bottom by headers or tees.

Four brine-pumps (the cold parts of which are inside the insulated chamber) are used to circulate the brine, two in general being used for freezing and two for chilling or attemperating brine. A warm brine-pump (thawing-pump), a brine attemperator and a steam-heated warm brine-tank (brine-heater) are also shown in the Figure. The scheme of brine circulation is explained under Fig. 27 (page 991) and Figs. 29 to 32 (pages 994–5).

A small and compact machine is fitted for the provision-room serving the passengers and crew on the outward voyage. This machine consists of an ammonia compressor driven direct by a single steam-cylinder, both of which are mounted on a box bed, one half of which contains the ammonia condenser and the other half the evaporator or brine-cooler, the brine connections being such that the provision-room on the homeward voyage can be cooled by the larger or cargo machine.

Fig. 29 shows diagrammatically the general principle of the brine distribution. It must be again pointed out that chilled meat or produce must be carried at a very steady temperature; and to attain this object the makers in the present case (1) circulate a large quantity of brine just below the desired temperature of the

FIG. 29.—*Diagram Arrangement of Attemperated Brine System.*
(The L. R. Co.)

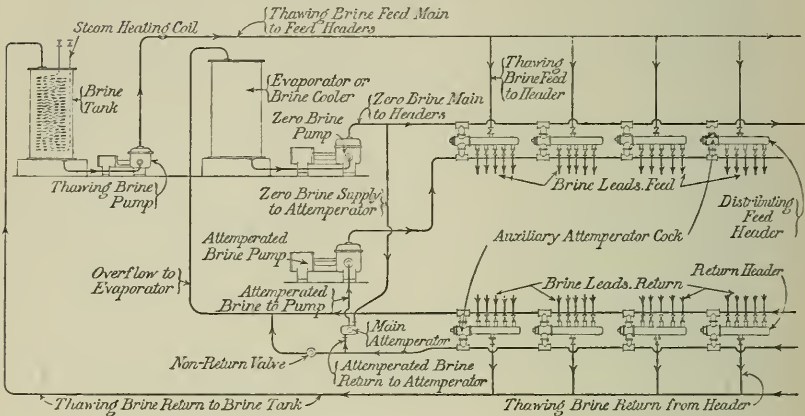



FIG. 30.—*Main Brine Attemperator.*

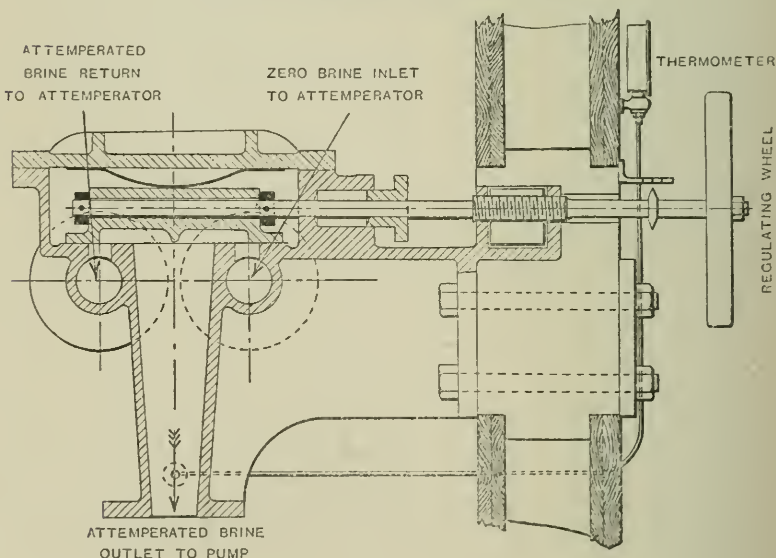


Fig. 31.—*Diagram Arrangement of Attemperated Brine System.* (The L. R. Co.)

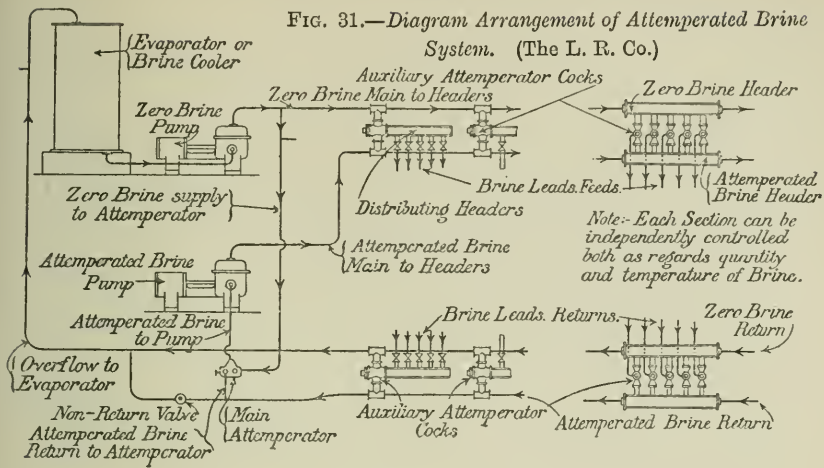
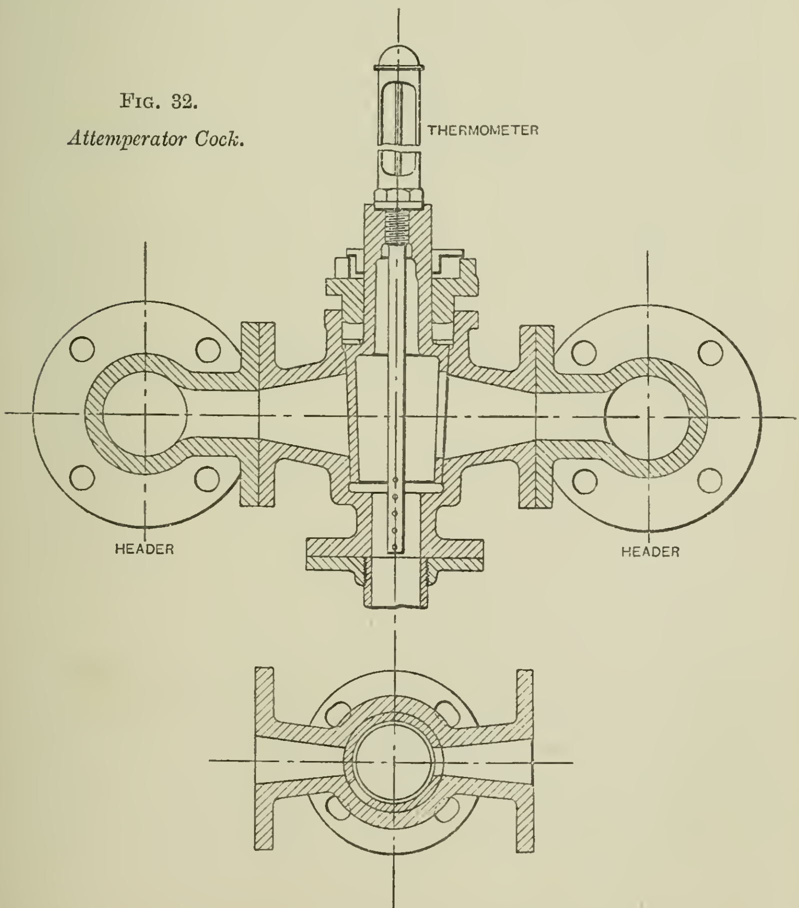



Fig. 32.

*Attemperator Cock.*



air in the chamber, rather than circulate a small quantity of brine at a much lower temperature, (2) to enable chilled and frozen produce to be carried economically by one machine without change of system or carrying evaporators working with different temperatures, and therefore with varying suction pressures.

In Fig. 29 the zero or freezing brine-pump draws directly from the evaporator and discharges brine at about 5° F. to a main which is connected with a series of "distributing feed-headers." Each of these headers supplies a number of "brine leads," or separate circuits leading to one hold. A second feed of attemperated brine, at about 35° to 38° F., is also taken to the headers, and a cock (auxiliary attemperator cock) is cross connected between the two feeds, and is so designed as to make it possible to circulate (1) all freezing brine through the leads; (2) all attemperated brine; (3) brine at any temperature between 5° and 38° F. (say). A corresponding cock on the return directs the brine into the main return most suited to its temperature.

The greater portion of the chilling brine flows back to the attemperator, where it is mixed with a little of the freezing brine, and then once more sent back to the holds as "chilling brine." The attemperator (Webb's or other designs) is a simple but effective device for mixing the highest returning brine (say 39° or 40° F.) with the lowest temperature brine (say 5° F.) in order to get a resultant brine of the right temperature (say 35° F.). A simple slide, Fig. 30, operates over two openings, one being the zero brine supply and the other the return attemperated brine, the combined areas of the openings always being a constant; if one is completely shut off the other must be full open. A hand-wheel and thermometer allow a perfect adjustment to be made.

The design of the auxiliary attemperator cock can be followed from the enlarged section in Fig. 32; the thermometer fitted allows each header to be supplied with brine at a pre-arranged temperature. An ordinary cock or valve is fitted to each lead or circuit, so that (1) any circuit can be cut out completely, if necessary; (2) a thermometer being fitted on each return circuit, the separate return cock enables the brine to be so regulated that the quantity best

suited for the work to be done by that circuit can be accurately adjusted.

The thawing brine arrangements are simple and are easily followed in the diagram. It can readily be understood, however, that instead of having a separate lead, as shown, one of the other mains can be used, inasmuch as no other temperature brine will be in circuit when the thawing-off brine is being used. A further possible arrangement of the auxiliary attemperator cocks is shown in the right-hand portion of Fig. 31, where the cock is placed between the two headers, and the brine lead or circuit is taken directly from the bottom.

The diagrammatical arrangements shown in Figs. 29 and 31 will enable the double set (forward and aft) of duplicated feeds followed on the Nelson boats to be easily understood. The two brine-distributing houses shown on Fig. 26 (page 991) are each supplied with freezing and chilling brine-feeds, as shown on Fig. 27, the main attemperator being fixed in the machine space. Auxiliary attemperator cocks are placed in the distributing houses and cross-connected to the two feeds in exactly the same way as shown in Fig. 29 (page 994). A header is attached to the bottom or outlet from which the separate circuits are fed into the holds. Corresponding return connections are made in the same distributing house, the main returns taking back the brine to the machinery space. The arrangement, which has given great satisfaction, was fitted by Messrs. The Liverpool Refrigeration Co. to the specifications of Mr. A. R. T. Woods, *Member*, the Superintendent Engineer for Messrs. Nelson.

*The Rating of Refrigerating Machines.*—The generally accepted units for the commercial rating of refrigerating machines are—

(1) The ice-melting capacity;

(2) The ice-making capacity,

both expressed in tons of 2,240 lb. per day of twenty-four hours.

*Ice-melting Capacity.*—A machine rated as "1 ton ice-melting capacity" would mean that under *assumed* or *named conditions as to range of temperature* the machine would remove the number of

thermal heat-units equivalent to that required to *melt* 1 ton of ice at 32° F. into water at 32° F.

The exact value of this unit has not been agreed upon by refrigerating engineers, the latent heat of water (fusion of ice) being variously taken as 142, 143, 143·7 and 144 B.Th.U. per lb., the possible values for the unit varying between 318,080 and 322,260 B.Th.U., according to the value selected for the latent heat of water. It is obvious, however, that in selecting any figure within the range given no actual unit for the 1-ton ice-melting capacity, or *ton of refrigeration* as it is more usually called, can be accepted which does not fix the temperatures between which the machine is supposed to work. Thus, a machine may be rated as $x$ tons of refrigeration when cooling water or milk and of $y$ tons when making ice, $x$ being greater than $y$, because the range of temperature in the machine will be less in the former case, and consequently the coefficient of performance, that is, $\frac{\tau_2}{\tau_1 - \tau_2}$, will be greater.

In the United States, 288,000 B.Th.U. per day per ton of refrigeration is the recognized figure, being made up of 144 B.Th.U. per lb. of ice melted and a ton of 2,000 lb. This gives 12,000 B.Th.U. per ton per hour and 200 B.Th.U. per ton per minute, the temperatures selected through which the machine is supposed to work being 90° F. in the condenser and 0° F. in the evaporator, and these would seem suitable temperatures for the States. No definite value has yet been accepted as a British standard, and it is suggested that 322,000 B.Th.U. per ton might be accepted, being the nearest round figure obtained by the product of 143·7 and 2,240, while a reasonable range of temperature for this country would be from 70° F. in the condenser to 0° F. in the evaporator.

*Ice-making Capacity.*—The ice-making capacity is a measure of the actual weight of ice made by a machine (designed for ice-making) in tons per twenty-four hours. It generally assumes the normal conditions of a British summer, the assumed ranges of temperature, as may be reasonably expected, varying with different makers.

Manufactured ice may be broadly divided into two kinds, *clear* and *opaque*, and these require varying refrigerating effects, but the difference is small compared with the large allowances necessary for the influx of external heat at the ice-tanks, evaporator, and through the pipe connections and the losses due to "thawing off," being as much as 70 per cent. for small machines (say up to 5 tons ice-making per day), and 25 per cent. for machines making 100 tons per day. Taking a typical example: supposing 1 ton of water at 63° F. is to be made into ice at 22° F., then approximately—

| | | | |
|---|---|---|---|
| 2,240 lb. water at 63° F. to water at 32° F. | = | 69,440 | B.Th.U. |
| 2,240 lb. water at 32° F. to ice at 32° F. . | = | 321,888 | ,, |
| 2,240 lb. ice at 32° F. to ice at 22° F. . | = | 11,200 | ,, |
| Total . . | | 402,528 | ,, |

This figure, together with the allowances mentioned, would become 700,000 B.Th.U. per ton of ice made for a machine of 5 tons ice-making capacity per day to 510,000 B.Th.U. per ton for a machine making 100 tons per day.

For a machine not designed for ice making, the unit of ice-making capacity would appear of little value, and the ice-melting unit should be used. Approximately the ice-making capacity is half the ice-melting capacity. As a fair practical rule, the author has found that

$$y = 0{\cdot}6\,x - 2,$$

where $x$ = ice-melting capacity in tons per day;

$y$ = ice-making capacity in tons per day.

*Conclusions.*—British refrigerating machinery by the leading makers is quite in keeping with the best traditions of the British engineer, but it would be idle to contend that the machines and their method of working are not capable of improvement. Vapour-compression machines have the merit of a wonderful simplicity and practically a standardization of design.

The two outstanding wants in refrigeration are:—

(1) A standard unit of refrigeration—say a "ton of refrigeration" of fixed value in B.Th.U., and taken between standard limits of temperature.

(2) A standard refrigerating machine of comparison.

Both these wants could be satisfied by a committee of this Institution in conference with the leading makers.

Careful research is also required to determine :—

(*a*) The relative efficiencies of the principal refrigerants at varying temperatures.

(*b*) The advantages and disadvantages of both wet and dry compression.

(*c*) The value of compound compression.

(*d*) The effect of varying the amount of refrigerant used in a given machine (undercharged and overcharged machines).

(*e*) The value of cooling the liquid refrigerant in (or by) the evaporator before expansion.

(*f*) Deterioration and fatigue of the refrigerant due to constant re-use. The presence of foreign vapours (such as water or oil) in the closed circuit. Value of by-pass driers or scrubbers.

The author's thanks are due to Messrs. L. Sterne and Co., Crown Iron Works, Glasgow; The Liverpool Refrigeration Co., Water Street, Liverpool; The Haslam Foundry and Engineering Co., Derby; and Messrs. Wm. Douglas and Sons, Putney, London, S.W., for supplying those drawings which, in the opinion of the author, best suited the scheme of the Paper, each drawing sent being by the firm at the author's special request. The above firms have also supplied photos which have greatly added to the value of the illustrations.

The Paper is illustrated by Plates 42 to 45 and 32 Figs. in the letterpress, and is accompanied by 6 Appendixes illustrated by 5 Figs.

[*The Discussion on this Paper was combined with that on the Paper by* Dr. J. H. Grindley, *and commences on page* 1054.]

# APPENDIX I.

## Notation.

*British Standard Units.*

H = quantity of heat, B.Th.U. per lb.
W = work, foot-pounds.
J = Joule's equivalent.
A = reciprocal of J.
$\tau$ = absolute temperature °F.
$t$ = temperature °F.
S = sensible heat, B.Th.U. per lb.
L = latent heat ($L_i$ internal, $L_e$ external), B.Th.U. per lb.
$p$ = pressure in pounds (absolute).
$v$ = specific volume of vapour, cubic feet.
$s$ = specific volume of liquid, cubic feet.
$c$ = specific heat of liquid.
$K_p$ = specific heat, constant pressure.
$K_v$ = specific heat, constant volume.
E = intrinsic energy.
$x$ = dryness fraction.
$i$ = enthalpy.
$\phi$ = entropy.
$\eta$ = coefficient of performance.

## APPENDIX II.

### Properties of Vapours and Refrigerants.

For a dry saturated vapour the heat, H, necessary to change its state from a liquid at 32° F. into dry saturated vapour at the temperature corresponding to its pressure is

$$H = S + L;$$

also
$$H = A\{E + p(v - s)\}, \quad . \quad . \quad \text{(a)}$$

where $p(v - s)$ is the external work, all the units being measured at or from 32° F.;

further
$$L = L_i + L_e,$$

and
$$L_e = A p (v - s)$$

from (a)
$$E = J H - p(v - s), \quad . \quad . \quad . \quad \text{(b)}$$

also
$$E = J(S + L_i)$$
$$= J(H - L_e).$$

For wet vapours or mixtures of liquid and vapours, if $x$ is the dryness fraction per unit of total weight,

$$H_1 = S_1 + x L_1,$$
$$H_1 = A\{E_1 + x p (v - s)\}, \quad . \quad . \quad \text{(c)}$$

$H_1$ being the heat required to form wet vapour from liquid at 32° F., $S_1$ and $L_1$ being measured at the saturation temperature.

For superheated vapours the additional heat necessary to raise the temperature from $\tau_1$ (saturation temperature) to $\tau_3$ is

$$K_p(\tau_3 - \tau_1).$$

Hence the total heat $H_3$ for a superheated vapour is given by

$$H_3 = S_1 + L_1 + K_p(\tau_3 - \tau_1),$$

where
$$S_1 = \int_{491 \cdot 6}^{\tau_1} c \, d\tau,$$

$c$ being the specific heat of the liquid.

If $K_p$ is not constant, but has a value that can be given in terms of $\tau_1$, let $K_p = \int(\tau)$, then

$$H_3 = \int_{491 \cdot 6}^{\tau_1} c \, d\tau + L_1 + \int_{\tau_1}^{\tau_2} \int(\tau) \, d\tau. \quad . \quad \text{(d)}$$

*Enthalpy.*—If we consider a unit weight of a substance in any condition and write

$$i = \mathrm{A}(\mathrm{E} + p\,v), \quad . \quad . \quad . \quad \text{(e)}$$

the function $i$ is called the enthalpy of the substance, and is a function of particular value in refrigeration.

Equations (a) and (e) differ only by the small quantity $\mathrm{A}\,p\,s$, which can be neglected in practical refrigeration; and in this case the enthalpy can be taken as a measure of the total heat required to form saturated or superheated vapour at constant pressure.

Hence we may write

$$\mathrm{H} = \mathrm{A}(\mathrm{E} + p\,v),$$

$$d\,\mathrm{H} = \mathrm{A}(d\,\mathrm{E} + p\,d\,v)$$

or $$\mathrm{J}\,d\,\mathrm{H} = d\,\mathrm{E} + p\,d\,v;$$

but $$d(p\,v) = p\,d\,v + v\,d\,p;$$

hence $$\mathrm{J}\,d\,\mathrm{H} = d\,\mathrm{E} + d\,(p\,v) - v\,d\,p$$

$$= \mathrm{J}\,d\,i - v\,d\,p. \quad . \quad . \quad . \quad \text{(f)}$$

If the pressure is constant during any operation,

$$d\,p = 0$$

and $$d\,\mathrm{H} = d\,i;$$

hence $$i_2 - i_1 = \mathrm{H}_2 - \mathrm{H}_1, \quad . \quad . \quad . \quad . \quad \text{(g)}$$

a result which renders a $\tau - i$ or $\phi - i$ diagram particularly useful.

If adiabatic compression is assumed in the compressor, the change of $i$ during the adiabatic compression alone is equal to the work done during the complete cycle of operations in the compressor.

For $$\mathrm{W} = \int_{p_2}^{p_1} v\,d\,p,$$

and from equation (f) $\mathrm{J}\,d\,\mathrm{H} = \mathrm{J}\,d\,i - v\,d\,p$, since for adiabatic operations $d\,\mathrm{H} = 0$,

$$\mathrm{J}\,d\,i = v\,d\,p,$$

or $$\mathrm{J}\,(i_1 - i_2) = \int_{p_2}^{p_1} v\,d\,p;$$

hence $$\mathrm{W} = \mathrm{J}\,(i_1 - i_2), \quad . \quad . \quad . \quad \text{(h)}$$

that is, the change of $i$ measured in work units will give the work done in the compressor per unit weight of fluid per cycle, on the

assumption that the compression is adiabatic—an assumption that does not lead to serious error in practical work.

*Enthalpy and the Free Expansion Operation.*—Consider a unit weight of refrigerant on the condenser side of the regulating valve, under condition $p_1$, $v_1$, $\tau_1$, $E_1$, no heat is received from or given to outside sources and no external work is done.

In passing the valve, the work done on the liquid is

$$v_1 (p_1 - p_2);$$

assuming $v_1$ constant for liquid,

work done by vapour $= p_2 (v_2 - v_1)$;

therefore, total work done

$$= p_2 (v_2 - v_1) - v_1 (p_1 - p_2)$$
$$= p_2 v_2 - p_1 v_1.$$

This work is done at the expense of the internal energy of the fluid.

Therefore $$E_2 = E_1 - (p_2 v_2 - p_1 v_1),$$
$$E_1 + p_1 v_1 = E_2 + p_2 v_2;$$

that is, $$i_1 = i_2.$$

The enthalpy of the fluid is, therefore, unchanged by the free expansion operation.

*Dry Saturated Vapours.*—Experimental results relating to dry saturated vapours lead to the following conclusions:—

(1) The temperature $\tau$ at which a liquid vaporizes depends solely on the pressure $p$ to which the liquid is subjected.

(2) The specific volume $v$ of the dry vapour depends on the temperature $\tau$ of vaporization.

(3) The sensible heat $s$ and latent heat L each depend on the temperature $\tau$ at which the liquid vaporizes.

In general, for any dry saturated vapour, if values for $\tau$, $s$, and either $v$ or L are determined experimentally, the remaining quantities H, L or $v$, $\phi$ and $i$ can be calculated.

*Refrigerants.*—For the fluids in general use in refrigeration, namely, anhydrous ammonia ($NH_3$), carbonic anhydride ($CO_2$), sulphurous anhydride or sulphur di-oxide ($SO_2$), it cannot be said that the figures available are beyond dispute, while complete $\tau - \phi$, $\tau - i$ and $\phi - i$ diagrams or charts would be of the utmost value.

*Anhydrous Ammonia* ($NH_3$).—For the dry vapour state the connection between the pressure and temperature has been determined by Regnault; recent researches, while differing somewhat, tend on the whole to confirm his figures. For temperatures below 32° F., the values of L and $v$ have been computed only. For temperatures higher than 32° F. the experiments of Dieterici have furnished values of the specific volume $v$ and $s$ of the dry vapour and liquid respectively, enabling L to be calculated and giving for the specific heat of the liquid

$$c = 1 \cdot 118 + 0 \cdot 001156\,(t - 32),$$

and $s$ can be obtained from

$$s = \int_0^t c\,d\,t.$$

From $c$ and L the values of $\phi$ of the vapour at different pressures can be determined.

$$\phi = \int_{\tau_2}^{\tau_1} c\frac{d\tau}{\tau}$$

$$= \int_{\tau_2}^{\tau_1} f(\tau)\frac{d\tau}{\tau}. \qquad \text{(h)}$$

If $\tau_1$ is the temperature at which the liquid vaporizes, then the entropy of the liquid at $\tau_1$ is given by the equation (h), writing $\tau = \tau_1$, the resulting entropy being denoted by $\phi_w$, being the total entropy in the liquid state.

To change the liquid at constant temperature $\tau$ from liquid into dry saturated vapour, the change of $\phi$ is given by

$$\phi = \int \frac{d\,\mathrm{H}}{\tau}$$

$$= \frac{1}{\tau_1}\int d\,\mathrm{H},$$

since $\tau = \tau_1$; and $\int d\mathrm{H}$ is equal to the whole heat taken in during evaporation, that is, latent heat L, so that

$$\phi = \frac{L_1}{\tau_1},$$

this $\phi$ being denoted in the Tables by $\phi_s$. Hence for a dry saturated vapour its entropy is given by the sum of $\phi_w$ and $\phi_s$.

*Properties of $NH_3$.*

| | |
|---|---|
| Critical temperature . . . | 266·0° F. |
| Critical pressure . . . . | 1624·0 lb. per sq. inch. |
| Specific volume of liquid . . . | 0·0256 cubic foot (mean). |
| Specific heat of liquid . . . | 1·02. |
| $K_p$ . . . . . . . . | 0·508. |
| $K_v$ . . . . . . . . | 0·393. |
| $\gamma$ . . . . . . . . | 1·29. |

*Carbonic Anhydride* ($CO_2$).—The properties of this vapour have been difficult to determine for the complete range of temperatures found in refrigeration, for, at the higher temperatures of this range the vapour is near its critical temperature, and throughout the entire range the pressures are relatively very great.

For temperatures above 32° F., Amagat (following others) experimentally determined the density of the dry saturated vapour and of the liquid, his results leading to the values of $v$ and $s$ given in the Table. Amagat also found experimentally the connection between the pressure $p$ and the temperature $t$ in the saturated vapour, and with the previous values of $v$ and $s$ the values of L and $\phi_s$ have been calculated, leading to values of $p$, $t$, L and $\phi_s$ given in the Table.

The values of the specific heat $c$ of the liquid have not been experimentally determined, but Mollier has computed what may be regarded as reliable values for $c$, $s$ and $\phi_w$.

*Properties of $CO_2$.*

| | |
|---|---|
| Critical temperature . . . . | 88·43° F. |
| Critical pressure . . . . . | 1,071 lb. per sq. inch. |
| Specific heat of liquid . . . . | 0·98 (mean). |
| $K_p$ . . . . . . . . | 0·217. |
| $K_v$ . . . . . . . . | 0·171. |
| $\gamma$ . . . . . . . . | 1·26. |

TABLE 1.*—*Saturated Vapour of Anhydrous Ammonia* ($NH_3$).

(*Dieterici and Wobsa.*)

| Temperature $t$ Fahr. | Pressure lb. per sq. in. $p$. | Specific Volume of Vapour $v$. | Sensible heat of liquid S. | Latent Heat. | | | Entropy of the Liquid $\phi_w$. | $\frac{L}{\tau}$ $\phi_s$. | Temperature $t$ Fahr. |
|---|---|---|---|---|---|---|---|---|---|
| | | | | Total L. | External $L_e$. | Internal $L_i$. | | | |
| −22 | 16·93 | 15·81 | −58·88 | 590·3 | 49·5 | 540·8 | −0·1265 | 1·350 | −22 |
| −13 | 21·47 | 12·66 | −49·34 | 584·0 | 50·2 | 533·8 | −0·1048 | 1·309 | −13 |
| − 4 | 26·98 | 10·21 | −39·70 | 577·3 | 50·9 | 526·4 | −0·0835 | 1·269 | − 4 |
| + 5 | 33·61 | 8·30 | −29·88 | 570·4 | 51·5 | 588·9 | −0·0622 | 1·230 | + 5 |
| 14 | 41·51 | 6·81 | −20·06 | 563·5 | 52·1 | 511·4 | −0·0414 | 1·190 | 14 |
| 23 | 50·82 | 5·63 | −10·02 | 555·9 | 52·5 | 503·4 | −0·0206 | 1·152 | 23 |
| 32 | 61·73 | 4·66 | 0 | 547·9 | 53·1 | 494·8 | 0 | 1·116 | 32 |
| 41 | 74·44 | 3·91 | +10·15 | 539·7 | 53·5 | 486·2 | 0·0205 | 1·079 | 41 |
| 50 | 89·05 | 3·29 | 20·45 | 531·4 | 53·8 | 477·6 | 0·0405 | 1·042 | 50 |
| 59 | 105·80 | 2·79 | 30·81 | 522·2 | 54·0 | 468·2 | 0·0607 | 1·006 | 59 |
| 68 | 124·85 | 2·37 | 41·34 | 512·6 | 54·2 | 458·4 | 0·0805 | 0·971 | 68 |
| 77 | 146·26 | 2·04 | 51·89 | 502·7 | 54·3 | 448·4 | 0·1003 | 0·936 | 77 |
| 86 | 170·53 | 1·75 | 62·55 | 492·4 | 54·2 | 438·2 | 0·1207 | 0·901 | 86 |
| 95 | 197·47 | 1·51 | 73·42 | 481·5 | 54·1 | 427·4 | 0·1392 | 0·869 | 95 |
| 104 | 227·36 | 1·30 | 83·19 | 471·2 | 53·7 | 417·5 | 0·1583 | 0·834 | 104 |

* Tables 1, 2, and 3 have been compiled by Emile Gouault, Sub-Lieutenant in the French Navy (Ingénieur frigoriste), for the International Association of Refrigeration. (Slightly corrected.)

TABLE 2.

*Saturated Vapour of Carbonic Anhydride* ($CO_2$).

(*Amagat and Mollier.*)

| Temperature $t$ Fahr. | Pressure lb. per sq. in. $p$. | Specific Volume. Liquid $s$. | Specific Volume. Vapour $v$. | Sensible heat of liquid S. | Latent heat. Total L. | Latent heat. External $L_e$. | Latent heat. Internal $L_i$. | Entropy of Liquid $\phi_w$. | $\frac{L}{\tau}$ $\phi_s$. | Temperature $t$ Fahr. |
|---|---|---|---|---|---|---|---|---|---|---|
| −22 | 213·0 | 0·0155 | 0·4323 | −25·72 | 126·13 | 16·40 | 109·73 | −0·0533 | 0·2863 | −22 |
| −13 | 248·5 | 0·0157 | 0·3674 | −21·87 | 122·67 | 16·18 | 106·49 | −0·0448 | 0·2748 | −13 |
| − 4 | 288·3 | 0·0160 | 0·3132 | −17·87 | 118·86 | 15·88 | 102·98 | −0·0363 | 0·2611 | − 4 |
| + 5 | 333·7 | 0·0164 | 0·2674 | −13·73 | 114·71 | 15·53 | 99·18 | −0·0276 | 0·2470 | + 5 |
| 14 | 384·8 | 0·0167 | 0·2286 | − 9·38 | 110·12 | 15·08 | 95·04 | −0·0186 | 0·2326 | 14 |
| 23 | 440·2 | 0·0172 | 0·1952 | − 4·82 | 105·04 | 14·55 | 90·49 | −0·0097 | 0·2170 | 23 |
| 32 | 502·7 | 0·0176 | 0·1669 | 0·00 | 99·34 | 13·89 | 85·45 | 0·0000 | 0·2021 | 32 |
| 41 | 573·3 | 0·0182 | 0·1422 | + 5·17 | 92·91 | 13·15 | 79·76 | +0·0099 | 0·1857 | 41 |
| 50 | 648·9 | 0·0188 | 0·1205 | 10·76 | 85·54 | 12·23 | 73·31 | 0·0205 | 0·1679 | 50 |
| 59 | 732·7 | 0·0197 | 0·1010 | 17·01 | 76·84 | 11·09 | 65·75 | 0·0321 | 0·1482 | 59 |
| 68 | 825·0 | 0·0210 | 0·0840 | 24·21 | 66·15 | 9·61 | 56·54 | 0·0452 | 0·1255 | 68 |
| 77 | 928·7 | 0·0228 | 0·0672 | 33·19 | 51·51 | 7·23 | 44·28 | 0·0613 | 0·0968 | 77 |
| 86 | 1038·0 | 0·0268 | 0·0474 | 47·50 | 26·88 | 3·98 | 22·90 | 0·0868 | 0·0493 | 86 |
| 87·8 | 1060·7 | 0·0298 | 0·0412 | 53·77 | 15·11 | 2·22 | 12·89 | 0·0981 | 0·0275 | 87·8 |
| 88·4 | 1069·3 | 0·0346 | 0·0346 | 61·43 | 0·00 | 0·00 | 0·00 | 0·1120 | 0·0000 | 88·4 |

TABLE 3.

*Saturated Vapour of Sulphurous Anhydride* ($SO_2$).

(*Cailletet and Mathias.*)

| Temperature $t$ Fahr. | Pressure lb. per sq. in. $p$. | Specific Volume of Vapour $v$. | Sensible heat S. | Latent heat. | | | Entropy of Liquid $\phi_w$. | $\frac{L}{\tau}$ $\phi_s$. | Temperature $t$ Fahr. |
|---|---|---|---|---|---|---|---|---|---|
| | | | | Total L. | External $L_e$. | Internal $L_i$. | | | |
| −22 | 5·54 | 13·177 | −16·29 | 175·99 | 13·50 | 162·49 | −0·0351 | 0·4023 | −22 |
| −13 | 7·24 | 10·307 | −13·72 | 174·42 | 13·81 | 160·61 | −0·0293 | 0·3907 | −13 |
| − 4 | 9·23 | 8·223 | −11·07 | 172·66 | 14·04 | 158·62 | −0·0234 | 0·3791 | − 4 |
| + 5 | 11·79 | 6·668 | − 8·39 | 170·68 | 14·26 | 156·42 | −0·0176 | 0·3675 | + 5 |
| 14 | 14·77 | 5·290 | − 5·65 | 168·48 | 14·45 | 154·03 | −0·0117 | 0·3559 | 14 |
| 23 | 18·32 | 4·328 | − 2·84 | 166·08 | 14·62 | 151·46 | −0·0059 | 0·3443 | 23 |
| 32 | 22·44 | 3·574 | 0·00 | 163·48 | 14·76 | 148·72 | 0·0000 | 0·3327 | 32 |
| 41 | 27·40 | 2·950 | + 2·90 | 160·65 | 14·87 | 145·78 | +0·0059 | 0·3210 | 41 |
| 50 | 33·23 | 2·437 | 5·85 | 157·61 | 14·90 | 142·71 | 0·0117 | 0·3094 | 50 |
| 59 | 39·90 | 2·036 | 8·86 | 155·36 | 14·94 | 140·42 | 0·0176 | 0·2978 | 59 |
| 68 | 47·57 | 1·715 | 11·92 | 150·93 | 14·94 | 135·99 | 0·0234 | 0·2862 | 68 |
| 77 | 56·23 | 1·443 | 15·03 | 147·28 | 14·90 | 132·38 | 0·0293 | 0·2746 | 77 |
| 86 | 66·31 | 1·218 | 18·20 | 143·40 | 14·81 | 128·59 | 0·0351 | 0·2629 | 86 |
| 95 | 77·53 | 1·042 | 21·42 | 139·32 | 14·69 | 124·63 | 0·0410 | 0·2513 | 95 |
| 104 | 90·17 | 0·882 | 24·68 | 135·05 | 14·55 | 120·50 | 0·0468 | 0·2397 | 104 |

*Sulphurous Anhydride* ($SO_2$).—Regnault established the relationship between the temperature and pressure, and furnished the data for $p$ and $t$ given in the Table. The values of $v$ and $s$ have been given by the experiments of Cailletet and Mathias, and those of $c$ by Mathias. The value of $c$ enables $s$ to be calculated, while the values of $v$ and $s$ enable L to be determined.

Knowing S and L, H, $\phi_w$ and $\phi_s$ can be found, and in this way the figures given in the following Table have been obtained.

*Properties of $SO_2$.*

| | |
|---|---|
| Critical temperature . . . | 312·8° F. |
| Critical pressure . . . . | 1,159·6 lb. per sq. inch. |
| Specific volume of liquid . . . | 0·0112 cubic foot (mean). |
| Specific heat of liquid . . . | 0·40 (mean). |
| $K_p$ . . . . . . . | 0·154. |
| $K_v$ . . . . . . . | 0·123. |
| $\gamma$ . . . . . . . | 1·25. |

## APPENDIX III.

### General Theory of Compression Machines.

The function of a refrigerating machine being to pump out or remove heat, the merit or "coefficient of performance" of such a machine is given by the ratio of the heat equivalent of work expended to the heat removed, or

$$\eta = \frac{H_2}{AW} \cdot \qquad \text{(k)}$$

The ideal refrigerating machine or heat pump is a reversed heat engine working in the Carnot cycle. The "coefficient of performance" therefore being

$$\eta = \frac{\tau_2}{\tau_1 - \tau_2}, \qquad \text{(l)}$$

on a $\tau - \phi$ diagram (Fig. 33)

$$\eta = \frac{\text{Area DCMN}}{\text{Area ABCD}}$$

$$= \frac{\text{ND}}{\text{DA}}$$

$$= \frac{\tau_2}{\tau_1 - \tau_2}.$$

*Work Done in the Compressor.*—The cycle of a vapour-compression machine is the reverse of that occurring in an ordinary steam-engine, and the diagram traced is that of the Rankine cycle reversed.

Fig. 33.

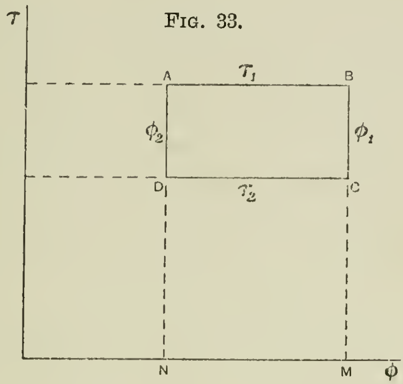
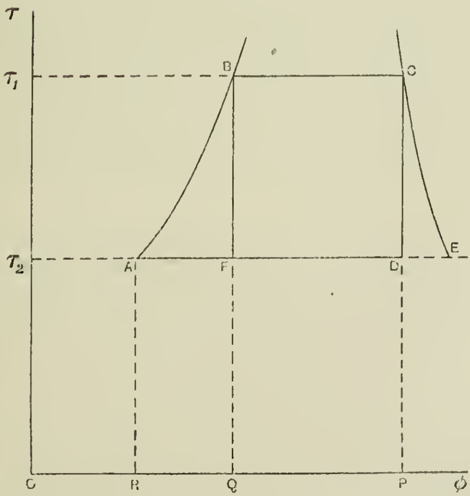



Fig. 34.




Taking first the Rankine cycle, let A, Fig. 34, represent the $\tau - \phi$ conditions of a unit mass of the liquid. Allow it to change

its state at constant pressure from liquid at $\tau_2$ to dry saturated vapour at $\tau_1$.

Work is now done in a cylinder, expansion being assumed adiabatic down to the temperature $\tau_2$.

By condensation the vapour can now be brought to a liquid at $\tau_2, p_2$ to its original state at A. In Fig. 34 the area under A B represents the heat absorbed during the operation A B, namely, the increase in the sensible heat of the liquid, so that the area $ABQR = S_1 - S_2$. Also the area under B C gives the latent heat L, taken in during the change of state from B to C, while the area under D A gives the heat rejected to the condenser. The whole heat transferred to work is given by the area A B C D.

Assuming no superheating occurs in the compressor of a refrigerating machine, and following the operations in the reverse order, namely, evaporation A D, adiabatic compression D C, condensation C B, and further cooling to bring the refrigerant to its original state A, the area A B C D now gives the work done by the compressor on the fluid.

If it were a practical proposition to fit an expansion cylinder in place of the free expansion through the regulating valve, the work represented in heat units by the triangular area A B F would be restored by the refrigerant, and the net work done would be given by the area F B C D, which represents a Carnot cycle.

*Refrigerating Effect.*—In an ideal vapour-compression machine cycle, the refrigerating effect is given by the area under F D, which is equal to

$$\tau_2 \,.\, FD = \tau_2 \,.\, BC,$$

that is,

$$\tau_2 \frac{L_1}{\tau_1}. \quad . \quad . \quad . \quad . \quad (m)$$

With, however, the free expansion, as adopted in practice, the effect is reduced to M D P Q, Fig. 35, where the area F M Q R is equal to A B F in the ideal machine.

In practice these areas are only approximately equal, as will be indicated.

The enthalpy is unchanged by the free expansion operation, that is,

$$i_1 = i_2, \quad . \quad . \quad . \text{(Appendix II)}$$

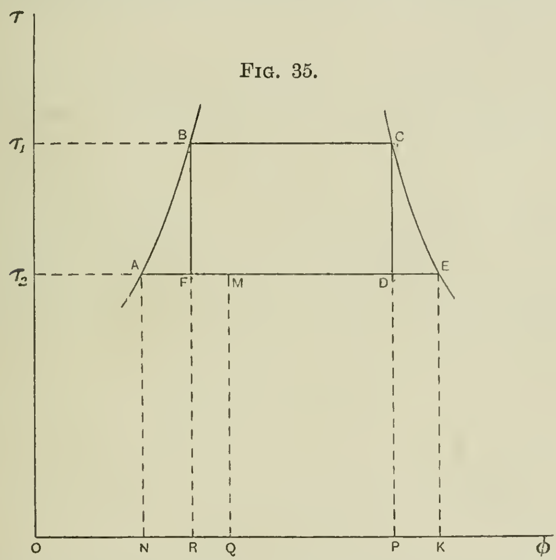
and further $i$ is *practically* the same as the heat supply at constant pressure, that is,

$$i_1 = S_1 + Ap_1s,$$

for $S_1$ is the total heat given to the liquid at pressure $p_1$, to bring it into the condition shown at B, Fig. 35, at temperature $\tau_1$ from liquid at 32° F.

The cooling of the fluid from B to A is due to a portion of the

Fig. 35.




liquid vaporizing on passing the regulating valve, the heat required to do this being absorbed from the wet liquid at B, which is thereby cooled to A. As this cooling is done by internal means, it follows that, not only is the *expansion work lost* but its equivalent in cooling has to be supplied from the fluid itself, and the *refrigerating effect is thereby further reduced.* This matter is a very important one in refrigeration and must be further considered, but it can be noted at once that the steeper the line A B the less the loss will be, while the longer the line A D the greater the refrigerating effect. The slope of the line A B depends on the specific heat of the liquid—a

low specific heat meaning a greater slope, while the length of the line A D depends on the latent heat—a high latent heat meaning a long line.

In a good refrigerant, therefore, *the ratio of the latent heat of the liquid to the specific heat of the liquid should be large.*

Further, let $x_1$ be the dryness fraction of the fluid after passing the regulating valve, then

$$i_2 = (S_2 + x_1 L_2) + A p_2 s,$$

for $S_2 + x_1 L_2$ measures the heat given to the fluid to form vapour of dryness fraction $x_1$, under constant pressure $p_2$. Hence, equating $i_1$ and $i_2$, we obtain

$$S_1 - S_2 = x_1 L_2 - A(p_1 - p_2)s, \quad . \quad . \quad (n)$$

or if we neglect the relatively small quantity $A(p_1 - p_2)s$,

we get $$x_1 = \frac{S_1 - S_2}{L_2}, \quad . \quad . \quad . \quad (p)$$

which gives in Fig. 35 the value of the ratio $\frac{AM}{AE}$, and so fixes the point M.

The approximate equality of the areas A B F and F M Q R can be shown, since the area under A B or A B R N is equal to $S_1 - S_2$, the heat taken in from A to B. The area under A M

$$\begin{aligned} AN \,.\, AM &= \tau_2 \frac{x_1 L_2}{\tau_2} \\ &= x_1 L_2, \end{aligned}$$

so that by equation (p) the area A B R N is equal to the area A M Q N. Taking away the common area A F R N, we get

$$ABF = FMQR.$$

The equality of these areas can only be accepted in approximate calculations, as the area A B F represents the loss of refrigerating effect only on the assumption that

$$A(p_1 - p_2)s$$

is small compared with $S_1 - S_2$.

This assumption is justified when using $NH_3$ or $SO_2$, but the percentage error when dealing with $CO_2$ cannot be overlooked.

For example, with $CO_2$, taking $t_1 = 86°$ F. and $t_2 = 14°$ F., from the Tables (Table 2, page 1008)

$$S_1 - S_2 = 56 \cdot 88$$

and $$A(p_1 - p_2)s = 2\cdot 15;$$

an error of 4 per cent. would thus be introduced.

The consequences of the free expansion, to summarize, are twofold:—

(1) The balance of the work done per unit weight of the refrigerant per cycle has been increased.

(2) The refrigerating effect has been diminished by an approximately equal amount.

The consequent loss is not a constant for all refrigerants, and must be taken into account when comparisons are made (Appendix IV).

*Coefficient of Performance with Wet Compression.*—The coefficient of performance in any assumed case can be obtained as a ratio of two areas on the $\tau - \phi$ diagram, if such be available, or it may be calculated as follows:—

On Fig. 35 the coefficient of performance $\eta$ is given by the ratio of the area M D P Q to the area A B C D.

Let $x_2 = \frac{AD}{AE}$ be the dryness fraction at the beginning of the compression, then the entropy at D is

$$c \log_e \frac{\tau_2}{491\cdot 6} + \frac{x_2 L_2}{\tau_2},$$

and the entropy at C, assuming constant specific heat for the liquid,

$$c \log_e \frac{\tau_1}{491\cdot 6} + \frac{L_1}{\tau_1}.$$

But the entropy at C is equal to the entropy at D, therefore

$$c \log_e \frac{\tau_1}{491\cdot 6} + \frac{L_1}{\tau_1} = c \log_e \frac{\tau_2}{491\cdot 6} + \frac{x_2 L_2}{\tau_2},$$

which gives $$x = \frac{\tau_2}{L_2}\left\{c \log_e \frac{\tau_1}{\tau_2} + \frac{L_1}{\tau_1}\right\} \quad . \quad . \quad (q)$$

or, using Tables, $$x = \frac{\phi_{s_1} + (\phi_{w_1} - \phi_{w_2})}{\phi_{s_2}}.$$

Thus $x_2$ can be calculated. The value of $\eta$ can now be determined.

The area A B C D = area under A B C − area under A D

$$= (S_1 - S_2 + L_1) - x_2 L_2,$$

and the area M D P Q = area under A D − area under A M,

$$= x_2 L_2 - \text{area under A B}$$

$$= x_2 L_2 - (S_1 - S_2);$$

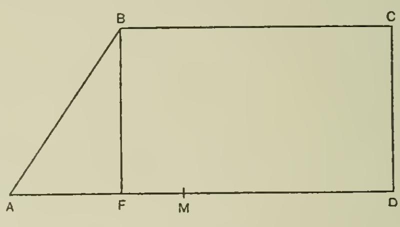
hence
$$\eta = \frac{x_2 L_2 - (S_1 - S_2)}{S_1 - S_2 + L_1 - x_2 L_2} \quad . \quad . \quad . \quad (r)$$

With $CO_2$ it is probably advisable to apply the small correction $- A(p_1 - p_2) s$ to the refrigerating effect.

Within the ordinary working temperatures of refrigerating machines no serious error is introduced by assuming the liquid line A B to be straight, and by the use of Tables the amount of calculation to determine $\eta$ can be reduced.

Thus, in Fig. 36 (the essential part of Fig. 35),

Fig. 36.




the refrigerating effect = area under M D

= area under F D − area under F M;

area under F D = $\tau_2$ . F D = $\tau_2$ . B C = $\tau_2 \phi_{s_1}$;

area under F M = area A B F

= $\frac{1}{2}$ A F . F B

= $\frac{1}{2}(\phi_{w_1} - \phi_{w_2})(\tau_1 - \tau_2)$.

Further, the area A B C D = the work done in heat-units

= A B F + B C D F;

area B C D F = $\phi_{s_1}(\tau_1 - \tau_2)$.

Thus all the elements necessary to find $\eta$ are determined.

*Example.*—In an ammonia compression machine, assuming $t_1 = 68°$ F. and $t_2 = 14°$ F., to find $\eta$ with wet compression.

From Table, Appendix I,

$$\phi_{s_1} = 0 \cdot 971, \quad \phi_{w_1} = 0 \cdot 080, \quad \phi_{w_2} = -0 \cdot 041,$$

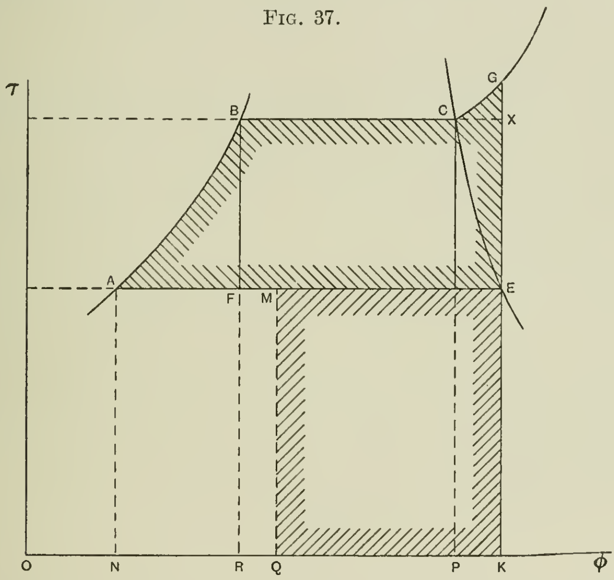
$$\tau_1 = 68 + 460 \text{ (approx.)} = 528°, \text{ and } \tau_2 = 474°,$$

$$\eta = 8 \cdot 2.$$

*Dry Compression.*—If the process of evaporation had been complete before compression commenced, the adiabatic operation would be represented by E G, Fig. 37, and the temperature $\tau_3$ at the end of compression would be found most easily from the diagram, by finding the intersection of the adiabatic E G with the constant pressure ($p_1$) line A B C G. The vapour is then superheated by an amount $\tau_3 - \tau_1$.

The direct effect is to increase the work done by an amount

Fig. 37.




represented by the area below C G as far as the horizontal at E, and to increase the refrigerating effect by the area thence to K P. From a theoretical standpoint, this results in a loss.

*Coefficient of Performance using Dry Compression.*—The coefficient of performance may be obtained by measurement of areas on the $\tau - \phi$ diagram, if available; otherwise, by calculation, since the area C G K P is equal to $K_p (\tau_3 - \tau_1)$, we require to know $\tau_3$. This is given directly by the fact that $\phi$ at G is equal to the $\phi$ at E; equating these values, we get

$$\phi_{w_1} + \phi_{s_1} + K_p \log_e \frac{\tau_3}{\tau_1} = \phi_{w_2} + \phi_{s_2}, \quad . \quad (s)$$

which suffices to determine $\tau_3$ if $K_p$ is known and the Tables of $\phi$ are available.

The area M E K Q, which gives the maximum refrigerating effect possible with dry compression, may be given in the form

$$L_2 - (S_1 - S_2),$$

and $$\eta = \frac{L_2 - (S_1 - S_2)}{S_1 - S_2 + L_1 - L_2 + K_p(\tau_3 - \tau_1)}, \quad . \quad . \quad (t)$$

an equation of doubtful value unless the quantities involved can be accepted without question.

A further method is to determine the work done AW directly from the $\tau - \phi$ diagram, or, by assuming the liquid line A B to be straight, proceeding as in the following example.

Assuming $t_1 = 68°$ F. and $t_2 = 14°$ F., find $\eta$ in an ammonia machine using dry compression.

From Table 1 (page 1007),

$$\phi_{s_1} = 0 \cdot 971 \qquad \phi_{w_1} = 0 \cdot 080$$
$$\phi_{s_2} = 1 \cdot 190 \qquad \phi_{w_2} = -\ 0 \cdot 041$$

Taking $K_p = 0 \cdot 508$, we obtain from equation (s) (or from $\tau - \phi$ diagram) that $\tau_3 = 682°$ F. (approx.).

Determining the equivalent areas to A B F, B X E F and C G X, in Fig. 37, on a $\tau - \phi$ diagram, we obtain

$$\eta = \frac{518 \cdot 9}{72 \cdot 92} = 7 \cdot 12.$$

## APPENDIX IV.

### Theoretical Comparison of $NH_3$, $CO_2$, and $SO_2$, as Refrigerants.

*Thermodynamic Losses peculiar to the Refrigerants.*—In an ideal machine, working in a reversed Carnot cycle, it does not matter, thermodynamically, what substance is used as a refrigerant. It has been shown, in Appendix III, that in a practical machine, even with an expansion cylinder, there is an unavoidable loss represented by the area under A F, Fig. 37 (page 1017). With a regulating valve, such as obtains in practice, the loss is represented by the area under A M, Fig. 37.

This loss varies with different refrigerants, but must be held as telling against them as refrigerants.

The following Table shows the percentage loss of refrigerating effect (that is, $\frac{AM}{AE} \times 100$, in Fig. 37), when the upper and lower temperatures are 68° F. and 14° F. respectively.

| Refrigerant. | Latent heat $L_2$. | Liquid heat $S_1 - S_2$. | Refrigerating effect. | Percentage loss $\frac{(S_1 - S_2)\ 100}{L_2}$. |
|---|---|---|---|---|
| $CO_2$ | 110·65 | 32·08 | 78·57 | 29·0 |
| $NH_3$ | 577·4 | 58·50 | 518·9 | 10·1 |
| $SO_2$ | 168·18 | 17·27 | 150·9 | 10·28 |

This loss tells very heavily against $CO_2$. If we had taken the upper temperature limit at 86° F. instead of 68° F. the comparison would have been still more unfavourable to $CO_2$.

*Regulating Valve.*—The use of the regulating valve in place of an expansion cylinder must now be considered.

Referring to Fig. 37 (page 1017), the area under F E represents

the refrigerating effect with the expansion cylinder, and the area under M E the refrigerating effect when a regulating valve is used.

$$\begin{aligned}\text{Area under F E} &= \text{area under A E} - \text{area under A F};\\ &= L_2 - \tau_2 (\phi_{w_1} - \phi_{w_2});\\ \text{area under M E} &= \text{area under A E} - \text{area under A M};\\ &= L_2 - (S_1 - S_2).\end{aligned}$$

Using Tables and taking the same temperatures as before, viz. 68° F. and 14° F., the following Table is obtained :—

| Refrigerant. | $L_2 - \tau_2 (\phi_{w_1} - \phi_{w_2})$. | $L - (S_1 - S_2)$. | Difference. | Percentage difference. |
|---|---|---|---|---|
| $CO_2$ | 80·44 | 78·57 | 1·87 | 2·33 |
| $NH_3$ | 522·19 | 518·90 | 3·29 | 0·63 |
| $SO_2$ | 151·85 | 150·91 | 0·94 | 0·62 |

Admitting that the use of the regulating valve is justified for practical reasons, the figures just given would seem to indicate that its use is more than justified, but it must be remembered that the effect of the substitution of the regulating valve for the impossible expansion cylinder on the coefficient of performance is more serious, for the heat representing the loss of refrigerating effect in the last Table is equivalent to the increase in the expenditure of work, since the area F M Q R is equal to the area A B F, Fig. 37 (page 1017).

With $t_1$ = 68° F. and $t_2$ = 14° F., as before :—

| Refrigerant. | Percentage loss by free expansion. | Calculated $\eta$. | Carnot cycle $\eta$. | Percentage loss from Carnot's cycle. |
|---|---|---|---|---|
| $CO_2$ | 17·4 | 6·70 | 8·78 | 24·7 |
| $NH_3$ | 5·1 | 7·12 | 8·78 | 18·8 |
| $SO_2$ | 5·3 | 7·47 | 8·78 | 14·9 |

## APPENDIX V.

### Outline Design of an Ammonia Compression Machine.

Assuming average temperatures of 70° F. and 0° F. for the ammonia in the condenser and evaporator, and allowing for losses at the regulating valve, influx of heat into the pipe connections and cold parts, and by superheating in the compressor, 450 B.Th.U. per pound of ammonia circulated may be taken as available refrigerating effect.

Taking one ton of refrigeration as 322,000 B.Th.U. per day of twenty-four hours, then

$$\frac{322{,}000}{450} = \text{lb. of ammonia to be circulated per day per ton,}$$

or

$$\frac{322{,}000}{450 \times 24 \times 60} \text{ lb. per minute.}$$

Taking the volume of 1 lb. of ammonia vapour at 0° F. to be 9·1 cubic feet, then

$$\frac{322{,}000 \times 9{\cdot}1}{450 \times 24 \times 60} = \text{cubic feet of ammonia vapour to be circulated per minute.}$$

$$= 4{\cdot}53 \text{ cubic feet of compressor displacement per minute per ton of refrigeration per day.}$$

This displacement would be required for machines of 2 tons of ice-making capacity per day and under, but a little less than this figure may be allowed for machines making 5 tons of ice per day and over. The following Table (page 1022) may be taken as a typical example of the variations of allowance according to size.

This Table is intended to illustrate commercial conditions, and only 300,000 B.Th.U. is allowed for *effective* ice-melting capacity per ton, the remaining 22,000 B.Th.U. representing a reserve of about 7 per cent.

It should be noted that for a machine making 5 tons of ice per day 700,000 B.Th.U. per ton is allowed, while for 100 tons the

amount is only 510,000 B.Th.U. per ton, which leads to a somewhat corresponding decline in the compressor displacement per minute.

| No. of Machine.* | Per day of 24 hours. | | | Compressor displacement. | |
|---|---|---|---|---|---|
| | Ice making. | Ice melting. | Effective B.Th.U. removed. | Rate. Cubic feet per min. per ton refrigeration. | Cubic feet per minute. |
| | Tons. | Tons. | | | |
| 1 | 5 | 11·7 | 3,500,000 | 4·4 | 51·5 |
| 2 | 10 | 20·0 | 6,000,000 | 4·35 | 87 |
| 3 | 25 | 45·0 | 13,500,000 | 4·3 | 193·5 |
| 4 | 50 | 87·0 | 26,000,000 | 4·2 | 365 |
| 5 | 100 | 170·0 | 51,000,000 | 4·1 | 697 |

* For reference in this Appendix only.

*Piston Speed.*—Owing to the fact that most compressors depend on self-acting mushroom valves for both the suction and delivery, the piston speed is much below ordinary steam-engine practice.

The maximum speed is about 350 feet per minute—this speed being reached both in the case of the quick-revolution, single-acting, vertical compressors, as well as in the slow-running, horizontal, double-acting machines.

The revolutions per minute vary from 45 in the latter case to 200 in the former type.

*Relation of Diameter to Stroke.*—It is most important that the whole of the contents of the compressor should be swept out at each stroke. Unless oil (De la Vergne and Sterne systems) is used, this is practically impossible, and so, however small the clearance may be, the piston must be well on its return stroke before the entrapped gas can expand to the inferior pressure.

Thus (when oil is not used) the longer the stroke in relation to the diameter for a given capacity, the greater the efficiency.

A long stroke, however, means a long connecting-rod and long bed-plate, with a correspondingly increased cost all round, and so the following is found to be a reasonable ratio:—

$$\frac{l}{d_1} = 2 \text{ to } 2\cdot4,$$

where $l$ = length of stroke in inches,

$d_1$ = diameter of compressor in inches.

*Thickness of Metal in the Compressor.*—The thickness of metal for the compressor walls is determined by the fact that cast-iron (the metal invariably employed) is porous, and 1 inch to $1\frac{1}{8}$ inch is required to ensure good working with ammonia at the pressures generally employed.

With very close-grained iron and plain castings, such as would be used for vertical machines, the metal may be reduced to $\frac{7}{8}$ inch, but it is not advisable to go below this, however small the compressor may be, nor should any cast-iron used in the circuit be thinner than this.

For compressors above 8 inches to 9 inches diameter a liner may be fitted with advantage, and this, if the rule given for the ratio of stroke to diameter be followed, should not be less than $1\frac{1}{8}$ inch, and may, with a compressor 21 inches diameter and over, be made $1\frac{1}{2}$ inch thick, it being remembered that the liner, as a rule, bears only at each end, and consequently must be made sufficiently stiff (*see* Fig. 9, page 968).

Compressor and other castings may, with advantage, be subjected to a solution of sal-ammoniac (ammonium chloride) under a pressure of 200 lb. per square inch. This has the effect of closing the pores by rusting them up; if the casting has to be machined, the treatment is more effective if done after the first cut has been taken.

*Valves.*—The diameter of the suction-valve may be made a little *above*, and the delivery valve a little *below*, one-third the diameter of the compressor.

*Suction- and Discharge-Pipes.*—Little or no difference is made between the diameter of the suction- and discharge-pipes, and they may both follow the rule

$$d_2 = 0 \cdot 25\, d_1,$$

where $d_1$ is the diameter of the compressor and $d_2$ the internal diameter of the connecting pipes.

*Compressor-Rod.*—The rod should be made of piston-rod steel, that is, of tensile strength of 35 to 40 tons per square inch with at least 21 per cent. elongation over 6 inches.

Taking $f_t$, the allowable stress in the rod, as 2,500 lb. per square inch (to allow for bending, etc.), and an effective pressure of 156 lb. per square inch on the piston, we have

$$\frac{\pi}{4} d_2{}^2 f_t = \frac{\pi}{4} d_1{}^2 p$$

$$d_2 = d_1 \sqrt{\frac{p}{f}}$$

$$= 0 \cdot 25\, d_1,$$

where $d_1$ = diameter of compressor,

$d_2$ = diameter of rod.

*Horse-power required to drive Compressor.* — The mean effective pressure for an ammonia compressor working under conditions suited for ice-making may be taken as 60 lb. per sq. inch; from this the probable "indicated horse-power" of the compressor may be computed; by adding 25 per cent., the required i.h.p. of the engine may be found.

The following rule may be followed:—

$b = 2\,a + c,$

where $b$ = i.h.p. of engine;

$a$ = ice-making capacity of machine in tons per 24 hours;

$c$ = constant = 7 for machines of 10 tons and below, and 10 for machines above 10 tons per day.

*Condensers.*—The square feet of surface required does not seem to depend on the ordinary laws for the transmission of heat through metal walls, but 300 to 550 B.Th.U. may be transmitted per square foot of mean surface per hour.

Assuming condensing water inlet not to exceed 65° F.:

Then for each ton of ice-melting capacity (ton of refrigeration),

*For submerged.*

*Ample*: 60 to 70 running feet of $1\frac{1}{4}$-inch internal diameter pipe.

*Small*: 50 running feet of $1\frac{1}{4}$-inch internal diameter pipe.

*Atmospheric.*

*Ample*: 90 running feet of $1\frac{1}{4}$-inch pipe.

*Small*: 60 running feet of $1\frac{1}{4}$-inch pipe.

Pipes $\frac{5}{32}$ inch to $\frac{3}{16}$ inch thick.

It should be mentioned that the reason why the difference between the ample and small allowances are closer in the case of the submerged than in the atmospheric, is, that the conditions, under which the submerged work, are much more stable than those of the atmospheric condenser. Again, one of the reasons why such variations in practice exist is due to the fact that, in many installations, allowances are made for sections to be cut out of circuit for inspection, renewals or repairs, without seriously affecting the working of the machine.

The grids of coils for atmospheric condensers may be spaced 12 inches to 20 inches apart, 12 feet to 20 feet long and 8 feet to 12 feet high.

Note.—Taking the mean diameter of the pipes one running foot of 2-inch pipe = 1·33 foot of $1\frac{1}{2}$-inch pipe = 1·6 foot of $1\frac{1}{4}$-inch pipe = 2 feet of 1-inch pipe.

*Evaporators.*—The many and very varied forms and uses of evaporators, the variations in practice with regard to the difference in temperature between the ammonia and the surrounding medium, make it quite impossible in the scope of this Appendix to give even the barest outline for the design of an evaporator. Suffice it to say

TABLE 4.—*Compressors.*

| Number of Machine. | Per day of 24 hours. | | | I.H.P. of Engine. | Compressor (Dimensions in Inches). | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Ice Making. | Ice Melting. | Effective B.Th.U. removed. | | R.P.M. | Diameter and Stroke. | Suction Valves. Diameter. | Delivery Valves. Diameter. | Diameter of Pipes. | Rod. |
| | Tons. | Tons. | | | | | | | | |
| 1 | 5 | 11·7 | $3·5 \times 10^6$ | 17 | 70 to 80 | $7\frac{1}{4} \times 15$ | $2\frac{1}{2}$ | $2\frac{1}{4}$ | 2 | $1\frac{7}{8}$ |
| 2 | 10 | 20·0 | $6 \times 10^6$ | 27 | 65 to 75 | $9 \times 18$ | $3\frac{1}{8}$ | $2\frac{7}{8}$ | $2\frac{1}{2}$ | $2\frac{1}{4}$ |
| 3 | 25 | 45·0 | $13·5 \times 10^6$ | 60 | 60 to 70 | $12 \times 24$ | $4\frac{1}{4}$ | $3\frac{7}{8}$ | 3 | 3 |
| 4 | 50 | 87·0 | $26 \times 10^6$ | 110 | 55 to 65 | $15·5 \times 30$ | $5\frac{1}{2}$ | 5 | 4 | $3\frac{7}{8}$ |
| 5 | 100 | 170·0 | $51 \times 10^6$ | 210 | 50 to 60 | $21 \times 36$ | $7\frac{1}{2}$ | 7 | 5 or 6 | $5\frac{1}{4}$ |

NOTE.—(*a*) Number of machine is for reference in this Appendix only.
(*b*) The higher figures in revolutions to meet overloads.

## TABLE 5.

*Heat Units and Condensing Water. Condensing Water on at 55° F. and off 80° F.*

| Number of Machine. | B.Th.U. per 24 hours. | | | | Condensing Water. Gallons per hour. Submerged. | |
|---|---|---|---|---|---|---|
| | Removed from cold body or effective. | Allowance for leakage into Machine and Connections, etc. | Heat Equivalent of Work Expended. | Total Removed. | Total. | Per ton Ice Making. |
| 1 | 3,500,000 | 265,000 | 1,041,000 | 4,806,000 | 800 | 160 |
| 2 | 6,000,000 | 440,000 | 1,660,000 | 8.100,000 | 1,350 | 135 |
| 3 | 13,500,000 | 990,000 | 3,700,000 | 18,190,000 | 3,030 | 121 |
| 4 | 26,000,000 | 2,000,000 | 6,800,000 | 34,800,000 | 5,800 | 116 |
| 5 | 51,000,000 | 3,740,000 | 12,890,000 | 67,630,000 | 11,300 | 113 |

that if brine is being cooled under conditions similar to those which exist when ice is being made by the brine, then :—

For each ton of ice-melting capacity allow 125 to 150 running feet of $1\frac{1}{4}$-inch internal diameter pipe, it being understood that the greatest possible care is taken to ensure an efficient circulation of the brine.

Tables 4 and 5 summarize the preceding rules.

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## APPENDIX VI.

### Testing Vapour-Compression Refrigerating Machines.

The bewildering conflict of results of apparently reliable tests of vapour-compression machines, carried out, in many cases, by eminent authorities, more particularly in comparative and competitive tests between the three principal refrigerants in general use—$NH_3$, $CO_2$, and $SO_2$—shows conclusively that much study and careful research is required to put the knowledge of this class of machine on a satisfactory basis.

It must, however, be pointed out that published results of comparative tests nearly always lack essential figures. It is quite easy to make any one of the three types mentioned to come out best in a series of trials, and yet the records—published exactly as taken—to indicate that the machines were (apparently) working under exactly similar conditions.

Theoretically, it is best to keep the refrigerant saturated during its cycle (wet compression—*see* Appendix III, page 1010). The clearance in the compressor (that is, a practical consideration) is apt, however, to tell heavily against this system as compared with dry compression. The clearances in the compressor or compressors should always be included in the records.

Further, practice seems to indicate that the evaporator is more efficient under wet and the condenser under dry compression.

It follows, therefore, that possibly good results would be obtained by working the evaporator under wet conditions, and then thoroughly drying the vapour by some mechanical means (say, in a spiral chamber) in its passage from the evaporator to the compressor, thus superheating the vapour in the compressor to a limited extent, this in turn allowing the condenser to do its best duty. Records should therefore be given of the temperatures of the refrigerant at all important points in the circuit.

Both theoretically and practically the coefficient of performance of a machine is improved by cooling the liquid refrigerant, after it leaves the condenser and before it passes the regulating valve, to the lowest available temperature in the machine itself. If this is done directly or indirectly the records should show it.

In the following suggested methods for recording tests, it is assumed that the actual amount of energy supplied for driving the machine can be determined. Electric driving by a motor of known efficiency has much to recommend it.

Brine circulation in the evaporator is also assumed and for laboratory tests the brine can be heated electrically or by steam, a check being thus afforded on the heat eliminated by the machine. Endless possibilities (for laboratory work) are offered in the way of heat exchangers for utilizing the heat wasted in the condensing water, etc.

## Data Required.

*Compressor.*

Type ............................ {Double or single acting, horizontal or vertical. If fitted with water-jacket, extra records to be inserted accordingly.}

Diameter.............

Stroke.............

Clearance.—Back ............ Volume.............

Front ............ Volume.............

Diameter of compressor-rod.............

Volume swept through by compressor-piston per revolution............

*Condenser.*

Type..............

Diameter of pipe {Internal..............
External..............

No. of sections..............

Length of pipe in each section ..............ft.

Total length of pipe ..............ft.

Estimated heating surface {Internal ..............sq. ft.
External ..............sq. ft.

Material of pipe..................................

Remarks *re* circulation of water..................................

*Evaporator.*

Same as for condenser, and in addition—

Method (if any) for agitating the brine..................................

Insulated or in insulated space..............................

*Brine.*

Salt employed..............

Tables of specific heats and specific gravities of the brine for ranges of temperature used in the test......................................

Note.—It is better to run the machine for some hours before observations are taken, in order to avoid allowances, which otherwise must be made due to varying temperatures and consequently varying specific heats of the brine.

Methods of measuring and checking the quantities of brine circulated ......................................................

*Refrigerant.*

Outline description of method employed for measuring the quantity (weight) of refrigerant circulated.

*Water.*

Methods of measuring and checking the quantities of water circulated.

Remarks.—Quality of water (town supply, well, canal, sea-water, etc. Hard or soft). Note if the water-supply be heated to a stated temperature before use, etc.

*Regulating Valve.*

Outline description. Record of movement (if any) during the test.

*Temperatures.*

Detailed account of method or methods adopted for reading temperatures.

*General Remarks.*

Outline description of any particular fitting likely to affect the test, such as a drier between the evaporator and compressor.

Amount of refrigerant in the machine. Interval of time between charging and testing the machine. Precautions taken to eliminate air or other foreign gases from both the refrigerant and brine circuits.

---

## Observations Required.

Date ............. Atmospheric conditions..........

Number of trial .............

Duration ...................

*Compressor.*

(1) Indicated horse-power.............

(2) Heat equivalent of (1) .............B.Th.U.

(3) Vapour entering compressor. Temp........... Pressure..........

(4) ,, leaving ,, Temp........... Pressure..........

*Condenser.*

(5) Temperature of water, inlet............

(6) ,, ,, outlet.............

(7) ,, ,, difference (5) and (6).............

(8) Quantity of water circulated ...........

(9) Heat rejected by condenser .............B.Th.U.

(10) Vapour entering condenser. Temp........... Pressure..........

(11) Liquid leaving ,, Temp........... Pressure..........

*Evaporator.*

(12) Temperature of brine, inlet............

(13) ,, ,, outlet.............

(14) ,, ,, difference (12) and (13).............

(15) Quantity of brine circulated.............

(16) Allowance ± B.Th.U. for variations in (12) and (13) during trial ............

(17) Net refrigerating effect .............B.Th.U.

(18) Refrigerant entering evaporator. Temp......... Pressure........

(19) ,, leaving ,, Temp......... Pressure .......

*Heat Balance.*

(20) Net refrigerating effect (17) ..............B.Th.U.

(21) Heat equivalent of work expended in compressor (2) ..............B.Th.U.

(22) Total heat imparted to refrigerant (20) and (21) ..............B.Th.U.

(23) Total heat rejected at condenser (9) ..............B.Th.U.

*(24) Difference between (22) and (23) ..............B.Th.U.

*Efficiency.*

(25) Heat equivalent of energy supplied to machine ............B.Th.U.

(26) Coefficient of performance (*a*). Ratio of (2) to (17) ...............

(27) Coefficient of performance (*b*). Ratio of (25) to (17) ...............

(28) Efficiency of driving. Ratio of (25) to (2)...........................

(29) Capacity of machine. Ice melting per day of 24 hrs...............

*General.*

(30) Weight of refrigerant circulated .............lb.

(31) Estimated refrigerating effect from (30)..............

(32) Difference between (17) and (31) .............B.Th.U.

(33) Temperature of liquid refrigerant before passing the regulating valve.............

* This difference is generally fairly large. In commercial machines due allowances are made for "heat leakage" into the pipes and connections. *See* Table 5 (page 1027).

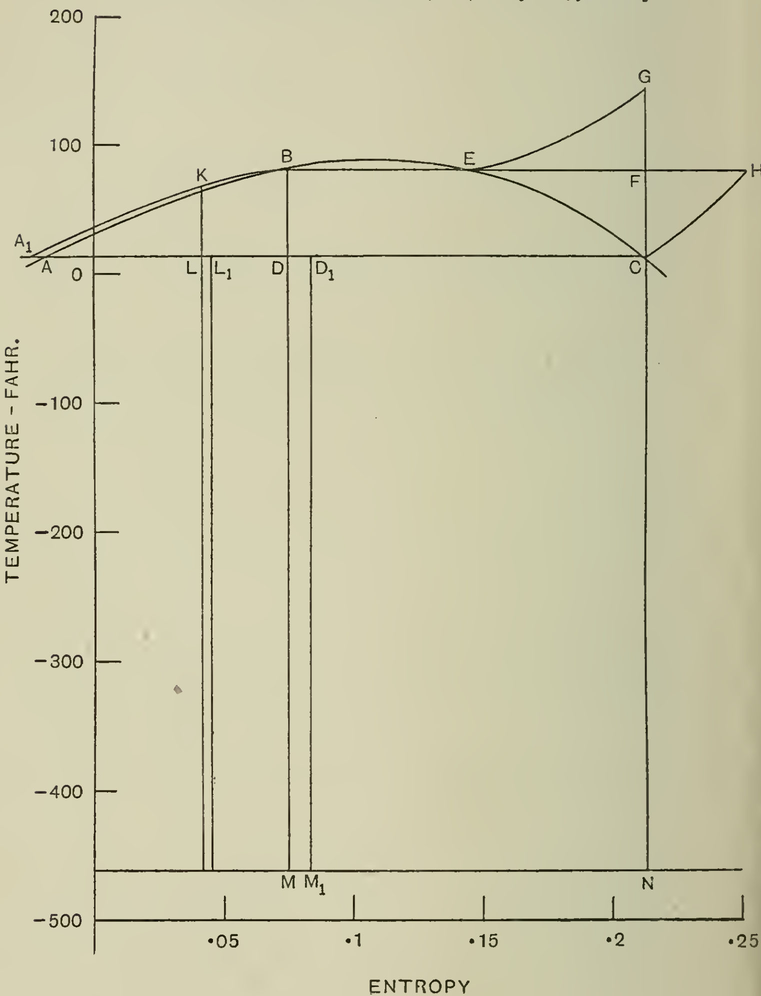
# A CONTRIBUTION TO THE THEORY OF REFRIGERATING MACHINES.

By JOHN H. GRINDLEY, D.Sc., *Member*, Principal, Crawford Municipal Technical Institute, of Cork.

*Introduction.*—In the author's study of Vapour-Compression Refrigerating Machines he has often been inconvenienced by the fact that the available diagrams, showing the heat properties of the refrigerants $CO_2$ and $NH_3$, are drawn to scales, the units of which are in the C.G.S. system and not in the, at present, more familiar British units, and the Tables giving the heat properties of the liquid and saturated vapours of these refrigerants sometimes show discontinuities which mar the conclusions made when discussing particular problems requiring a knowledge of differences between consecutive or near numbers in those Tables. Hence, since he desired to bring to the notice of refrigerating engineers a new cycle of operations in refrigerating machines, which would indicate that increased performances could be obtained from such machines, the author has included in his Paper certain Tables and diagrams which he has used in his work, English units being used throughout and no improbable discontinuities of moment appearing in the Tables.

Apart from these Tables and diagrams, the discussion from a theoretical standpoint of the new cycle of operations referred to

FIG. 38.—*Temperature-Entropy* ($\tau$ - $\phi$) *Diagram, for* $CO_2$.



forms the main part of this Paper, the advantages of the new cycle being clearly shown, especially when the temperature of the condensing water is high ; and though the cycle has not been tried in practice, so far as the author is aware, a discussion of its merits might lead to some conclusions of practical value.

*Description of a New Cycle of Operations.*—Fig. 38 shows a $\tau$-$\phi$ (temperature-entropy) diagram for $CO_2$ giving a particular cycle of operations for the refrigerant as follows :—Let ABC be the boundary line for the liquid and dry vapour of $CO_2$ drawn in the usual manner, and let the lines AC and BE be drawn at the lower and upper temperatures $t_1$ and $t_2$ at which the refrigerant takes in its heat and, in the main, rejects its heat respectively. Further, assume the compression to be dry and adiabatic.

Starting with the liquid in condition represented by the point B just before freely expanding it through the throttle, we should by using an expansion cylinder and expanding the liquid to the lower temperature adiabatically obtain the refrigerant in condition represented by D; but, owing to the abolition of the expansion cylinder, the actual condition of the refrigerant after free expansion is represented by $D_1$, the area under $DD_1$ ($DD_1M_1M$) being equal to the area $A_1BD$, the curve $A_1B$ being a constant-pressure line through B. The refrigerant now takes up heat represented by the area $D_1CNM_1$ until it becomes dry vapour represented by point C, and it is then compressed adiabatically as represented by CG, until the pressure reaches that given by the constant-pressure line EG drawn through E, after which it is condensed back to its original state represented by B.

This is the usual cycle and diagram discussed in the text-books, and it is there shown that the work done (W) in the compressor is represented (in heat units) by the area ABGCA and the refrigerating effect (R) by the area $D_1CNM_1$, the ratio $\frac{R}{W}$ of these two areas giving ($\eta$) the coefficient of performance.

It is a well-appreciated fact when using $CO_2$ that undercooling the liquid before freely expanding it shows a marked improvement in the performance of the machine, and it occurred to the author

that the following alterations in the cycle would show some gain in the performance.

The cycle of operations in actual machines being a continuous one, let the relatively hot liquid in condition represented by B, Fig. 38, before passing the throttle be passed through a narrow tube surrounded by a second larger tube through which the cold dry vapour in condition represented by C before compression passes, the liquid and vapour flowing in opposite directions so that the transfer of heat can be effected in as near as possible a regenerative manner. By doing this, the liquid could be cooled down considerably before being freely expanded, while the vapour would have become superheated to practically the upper temperature $t_2$. If the specific heats of the liquid and vapour were equal, the liquid could be practically cooled to the lower temperature $t_1$ while the dry vapour became superheated to $t_2$. In reality, owing to the difference in the specific heats, the dry vapour takes up heat as represented by the area under CH, Fig. 1, which is a constant-pressure line through C, and the liquid has been cooled to a temperature represented by K, where the whole area under BK is equal to the area under CH, one area representing the heat given up by the liquid and the other that taken in by the vapour.

Now, the condition of the gas being represented by H, let the cylinder be well jacketed by condensing water so as to produce compression as nearly as possible isothermal. The compression operation would then be represented by the constant temperature line HE, after which the vapour is condensed and returned to condition B, where in turn it could be undercooled to K.

The free expansion operation would then bring the liquid into condition represented by $L_1$, where area AKL = area under $LL_1$. The cycle of operations is then represented by $KL_1CHBK$. It involves :—

(1) Undercooling of the liquid before free expansion ;

(2) Superheating of the vapour before compression ;

(3) Isothermal compression.

Comparing this cycle with the usual cycle of operations, we find that the work done in the compressor (W) is represented for the

latter cycle by ACGBA and for the new cycle by ACHBA. The difference between these areas resolves itself into the difference between the triangular areas EFG and FCH, a difference obviously very small, and we may take the work done to be the same in the two cycles. As regards the refrigerating effect (R) the gain is at once obvious, the area under $L_1C$ representing the new refrigerating effect as against the area under $D_1C$ for the old cycle. The ratio $\frac{L_1D_1}{D_1C}$ represents the fractional increase in R and the area under $L_1D_1$ the net gain in R.

To give actual figures for the values of W and R in the two cycles, the author has calculated in B.Th.U. the numbers given in the following Table, the suffixes 1 and 2 distinguishing the old and new cycles respectively.

| Upper Pressure. | Temp. Limits. | | $W_1$. | $W_2$. | $R_1$. | $R_2$. | $\frac{R_1}{W_1}=\eta_1$. | $\frac{R_2}{W_2}=\eta_2$. | Per cent. Increase in Performance. |
|---|---|---|---|---|---|---|---|---|---|
| lb. per sq. in. | °F. | °F. | | | | | | | |
| 800 | 20 | 65·7 | 10·9 | 11·0 | 78·9 | 93·3 | 7·24 | 8·49 | 17 |
| 800 | 10 | 65·7 | 14·2 | 14·1 | 79·4 | 95·7 | 5·59 | 6·77 | 21 |
| 800 | 0 | 65·7 | 17·2 | 17·5 | 79·5 | 97·6 | 4·62 | 5·58 | 21 |
| 1000 | 20 | 82·5 | 15·0 | 14·5 | 59·4 | 79·5 | 3·96 | 5·48 | 38 |
| 1000 | 10 | 82·5 | 18·7 | 17·9 | 59·9 | 81·8 | 3·20 | 4·57 | 43 |
| 1000 | 0 | 82·5 | 21·3 | 21·5 | 60·0 | 83·6 | 2·75 | 3·89 | 41 |
| 1200 | 20 | 90·0 | 18·4 | 16·9 | 59·2 | 79·8 | 3·22 | 4·73 | 47 |
| 1200 | 10 | 90·0 | 22·4 | 20·2 | 59·7 | 81·9 | 2·66 | 4·06 | 53 |
| 1200 | 0 | 90·0 | 26·3 | 23·9 | 59·8 | 83·6 | 2·27 | 3·50 | 53 |

The figures in the last column lead to the conclusion that large increases in the performance of vapour-compression machines might be expected to follow from the adoption of the new cycle, especially

when using condensing water at high temperatures. The new cycle would therefore be expected to give the best improvements when used in machines working in hot countries.

For $NH_3$ machines an examination of the $\tau$-$\phi$ diagrams for the old and new cycles would show that increases in performance would also follow from the adoption of the new cycle, but they may be more difficult of realization in practice. Assuming the compressions to be strictly adiabatic or isothermal as the case may be, the calculations on the $\tau$-$\phi$ diagram would show a gain in performance sometimes as high as 20 per cent., part of which should undoubtedly be obtainable in practice.

The working of the proposed cycle involves two additions to the usual machines, first, an efficient water-jacket to the compression cylinder, so that the compression may be as nearly as possible isothermal, and second, a simple arrangement of inner and outer tubes as described, by which the cold vapour before compression can take up from the relatively hot liquid before passing through the expansion valve as much of its heat as possible, the transfer being accomplished in as nearly as possible a regenerative manner.

*Construction of the Tables for* $CO_2$ *and* $NH_3$.—The Notation used throughout and the units in which they are given in the Tables are as follows:—

$p$ = pressure on the refrigerant, lb. per square inch.
$t$ = temperature °F.
$\tau$ = absolute temperature °F. $\tau = 459\cdot6 + t$.
$s$ = specific volume of liquid, cubic feet.
$v$ = specific volume of dry vapour, cubic feet.
L = latent heat of evaporation, B.Th.U. per lb.
S = heat of the liquid, B.Th.U. per lb.
H = total heat of evaporation, B.Th.U. per lb.
$\phi$ = entropy.
$\phi_w$ = entropy of the liquid.

Two other functions are used, $i$ and $\psi$, which facilitate calculations on refrigerating machines, but which do not enter into the Tables, though they will be used later.

*Table for* $CO_2$.—The Table in Zeuner's Technical Thermodynamics * formed the basis of the interpolation work necessary for the new Table here given (page 1050), and the two Tables will be found to agree very closely, the figures in the new Table being given for every 5° F. The construction of the Table given by Zeuner is well described by him; the only point on which attention need be called is that the constants in Mollier's formula for L, namely, $L = a\,\tau^{0 \cdot 43} (b-\tau)^{0 \cdot 43}$ are slightly changed from the values given by him to $a = 1 \cdot 2264$ and $b = 548 \cdot 03$. Recent work seems to indicate that the values of L so obtained may be too small, but the evidence is too inconclusive to enable any corrections to be made even if necessary.

*Table for* $NH_3$.—The $NH_3$ Table given by Zeuner shows a discontinuity which is very inconvenient when making calculations. The trouble arises from the fact that two sets of data are available, and one of these, which happens to be the more reliable, applies only to temperatures above 32° F. After close examination of the figures and much calculation, the experiments of Dieterici on specific volumes and his calculations of L have been taken as the basis of the Tables for temperatures above 32° in the same manner that Zeuner adopted. At temperatures below 32° F. use has been made of Franklin and Kraus' result that at $- 27 \cdot 4°$ F. the latent heat of $NH_3$ was 613·8 B.Th.U. It was difficult to form a good connection between this value and the latent heats above 32° F., but since it is hardly likely that the latent heat would vary in such a manner as is shown by the older Tables, a formula like that suggested as an approximate one by Dieterici, namely, $L = a\sqrt{t_k - t}$ where $t_k$ is the critical temperature, should serve to connect the values. The only reasonable formula which gives results agreeing fairly well with Dieterici's figures above 32° F. and Franklin and Kraus' figure for L at $- 27 \cdot 4°$ was found to be

$$L = 40 \cdot 326\,(266 \cdot 9 - t)^{0 \cdot 48} \quad . \quad . \quad (1)$$

* English translation by Klein.

To illustrate these results the curves on Fig. 39 have been drawn: the line AB represents results obtained from (1), the points shown as crosses are obtained from Zeuner's Tables, and the point C represents Franklin and Kraus' result. When the scales on which this diagram is drawn, and the uncertainty attaching to any particular absolute value of L as plotted are considered, the better plan appeared to be to discard the values of L below 32° F. as

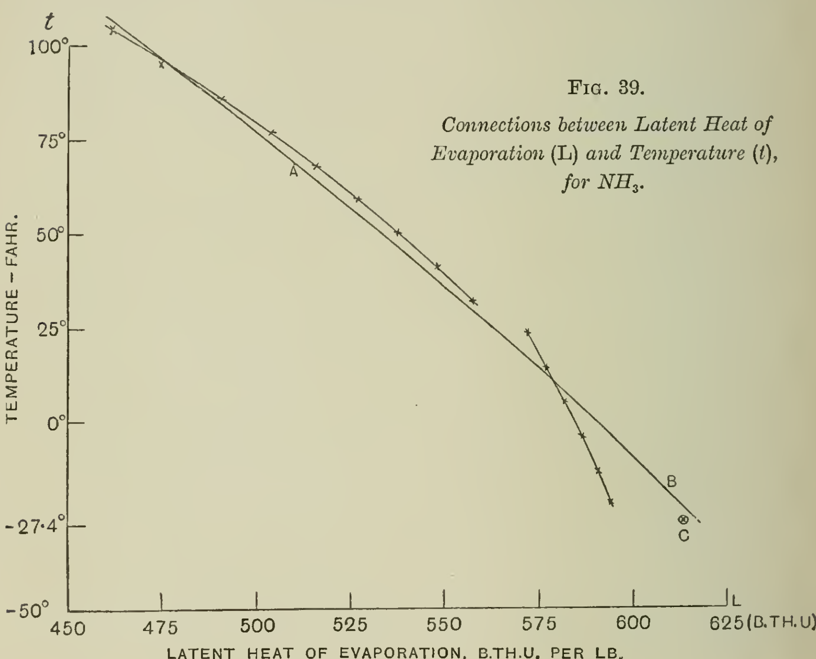



Fig. 39.

*Connections between Latent Heat of Evaporation* (L) *and Temperature* ($t$), *for* $NH_3$.

given by Zeuner and adopt the values given by (1), which have many points in their favour.

The pressure-temperature relation agrees with that given by Zeuner, the values of $p$ and $\tau$ being found to agree with those given by a formula of the type given by Nernst, namely, $p = \mathrm{A} + B \log \tau + \mathrm{C}\,\tau + \frac{\mathrm{D}}{\tau}$ with suitable values for the constants. The values of $\frac{dp}{dt}$ were obtained from this equation.

The specific volumes ($v$) have been calculated throughout from the values of L, by means of the well-known thermodynamic relation

$$\frac{dp}{dt}(v - s) = \frac{\mathrm{JL}}{\tau}$$

the values of $s$ being taken to vary uniformly from 0·0243 cubic foot at − 10° F. to 0·0281 cubic foot at 105° F. After much exhaustive calculation work, the above appears to be the best method.

Difficulties were experienced in calculating S, for the existing data are so very inconsistent. Above 32° F. Dieterici gives for the specific heat of liquid $NH_3$

$$c = 1 \cdot 118 + \cdot 001156\ (t - 32)$$

and this has been taken to hold for temperatures below 32° F., all other formulæ showing an improbably large variation of $c$ with $t$. From this equation we get for the heat of the liquid

$$\mathrm{S} = 1 \cdot 118\ (t - 32) + 0 \cdot 000578\ (t - 32)^2.$$

The calculation of the remaining columns in the Table (page 1051) offers no difficulty.

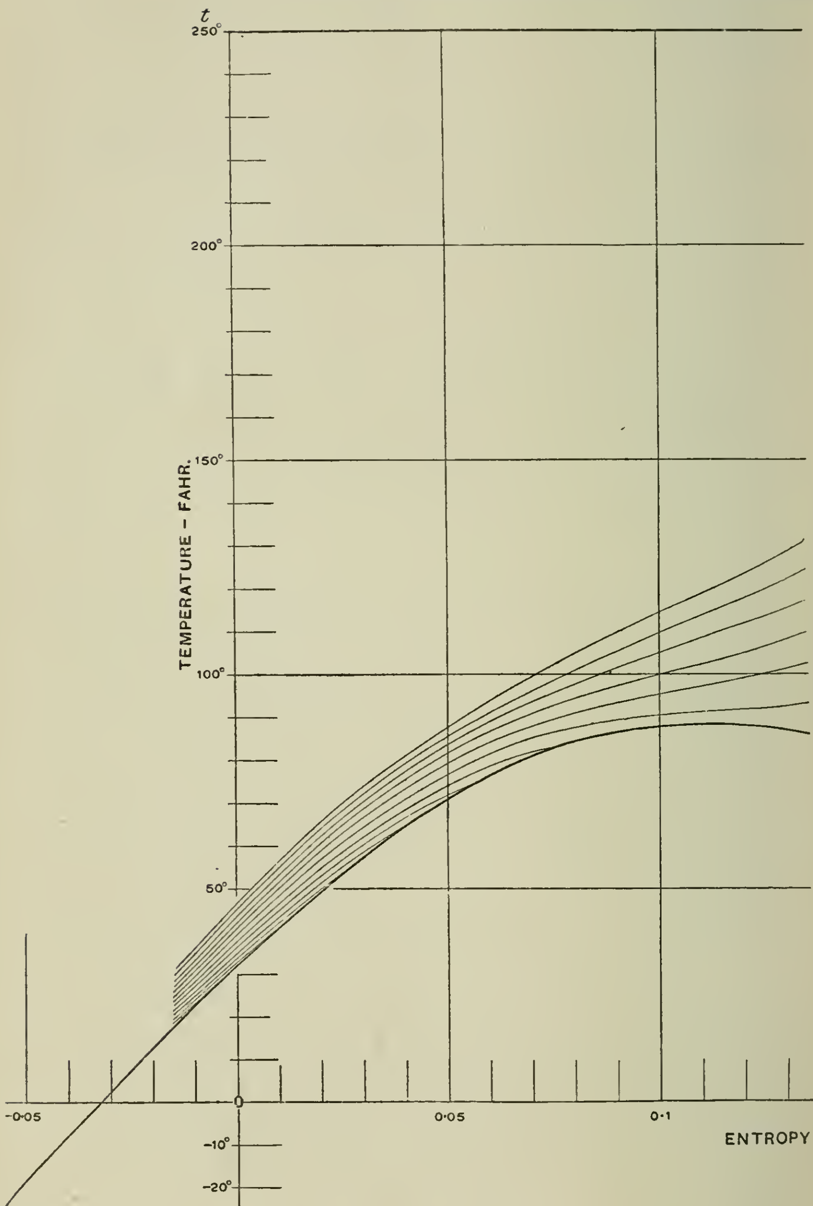
*On certain Diagrams which render easy Calculations on Refrigerating Machine Performances.*—The well-known $\tau$-$\phi$ diagrams have simple properties appreciated by engineers, for the work done in the compression cylinder and the refrigerating effects produced are represented by areas on these diagrams. A $\tau$-$\phi$ diagram for $CO_2$, using British units, is given in Fig. 40 (pp. 1042–3), the values of $\phi$ being obtained from Mollier's diagrams. The determination of areas is, however, not a simple matter if any calculations have to be made, and Mollier has drawn a diagram for $CO_2$, which has for co-ordinates $i$ and $\phi$, where $i$ is a function having the following properties :—*

1. Changes of $i$ in the refrigerant, while it gains or loses heat at constant pressure, measure directly the amounts of heat so gained or lost;
2. The change of $i$ in the refrigerant between the lower and upper limits of pressure in the compressor measures directly the work done in the compressor; and

* *See* Ewing's "Production of Cold," page 193.

FIG. 40.—*$\tau$ - $\phi$ Diagram for $CO_2$, the Values*

The work done in the compression-cylinder, and the refrigerating




(*Continued on next page.*)

*of $\phi$ being obtained from Mollier's Diagrams.*

effects produced are represented by areas on these diagrams.

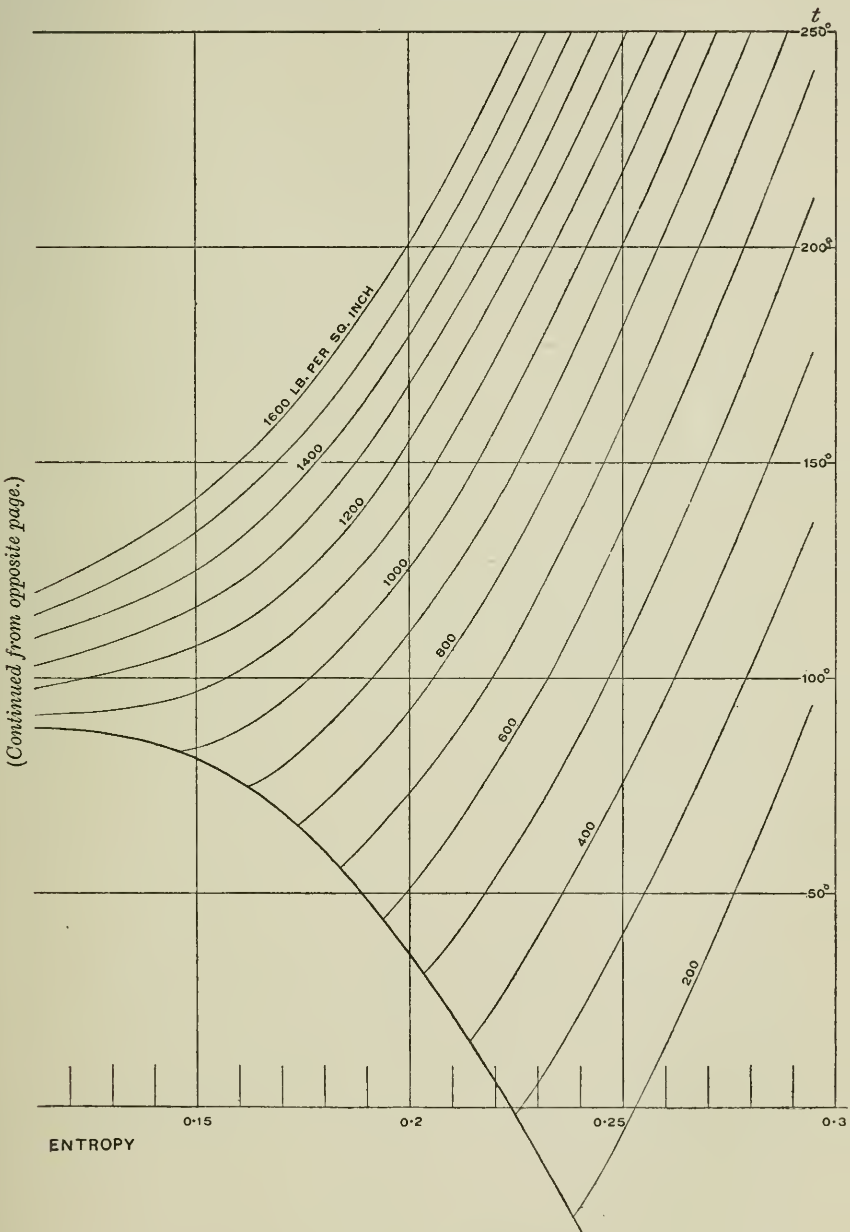
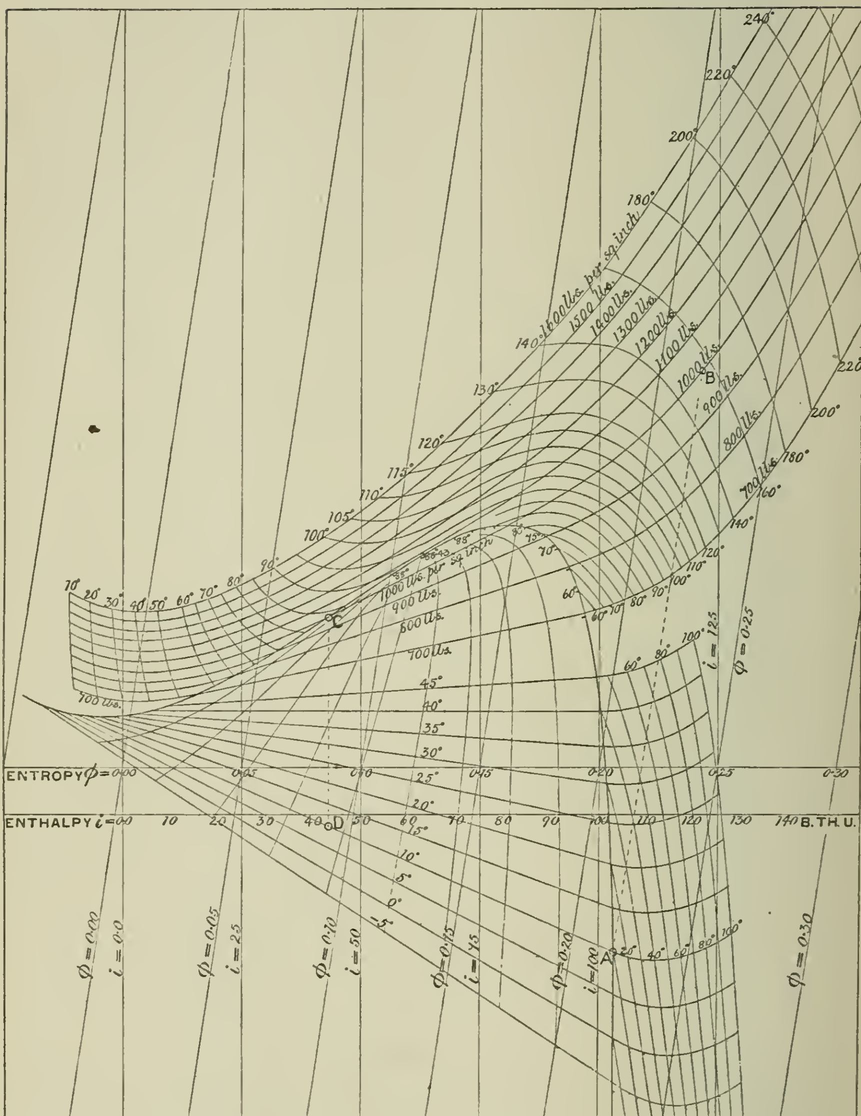


FIG. 41.—$\phi$ - $i$ *Diagram for* $CO_2$. After Mollier.



3. The value of $i$ remains unaltered by the free expansion operation through the throttle-valve.

Hence since the entropy $\phi$ remains constant during adiabatic operations, a diagram drawn with $\phi$ and $i$ for co-ordinates would give the work done and the refrigerating effect in any cycle using adiabatic compression as simple differences on the scale of $i$.

Unfortunately rectangular co-ordinates render the diagram very distorted, and use is made of oblique co-ordinates. Fig. 41 shows such a $\phi$-$i$ diagram using British units for the scales, and this diagram will not offer any inconsistencies with Mollier's, since it has been drawn by interpolation of the values of $\phi$ and $i$ given on Mollier's diagram.

To use the diagram, take for example the compression pressure to be 1,000 lb. per square inch, no undercooling, the lower temperature of the $CO_2$ as 10° F., and the compression adiabatic and dry. Then A on the boundary curve represents the condition of the dry vapour before compression. A line AB, parallel to the $\phi$ axis, to meet the constant-pressure line corresponding to the compression pressure represents the adiabatic compression. The gas is then condensed at constant pressure as represented by BC, where C is on the liquid boundary line, and a line CD drawn at right angles to the $i$ axis from C to the 10° F. line, gives the point D representing the condition of the refrigerant after free expansion, and the line DA represents the final operation of taking in heat. In calculations on W and R, all that is necessary is to find the value of the $i$ co-ordinate of each of the points A, B and C, for the value of $i$ at C and D are equal. If $i_A$, $i_B$ and $i_C$ are three such values, then $i_B - i_A = W$ and $i_A - i_C = R$. Numerically with the points taken, $i_A = 103 \cdot 1$, $i_B = 121 \cdot 5$, $i_C = 43 \cdot 2$, therefore $W = 18 \cdot 4$, and $R = 59 \cdot 9$ B.Th.U., and $\eta = 3 \cdot 26$.

Though such a diagram for $NH_3$ would not be quite so useful, since approximations can be made as to the dimensions of certain areas on the $\tau$-$\phi$ diagram, still for good work such a diagram would have its use, and the diagram on Fig. 42 (page 1046) is drawn for this purpose.

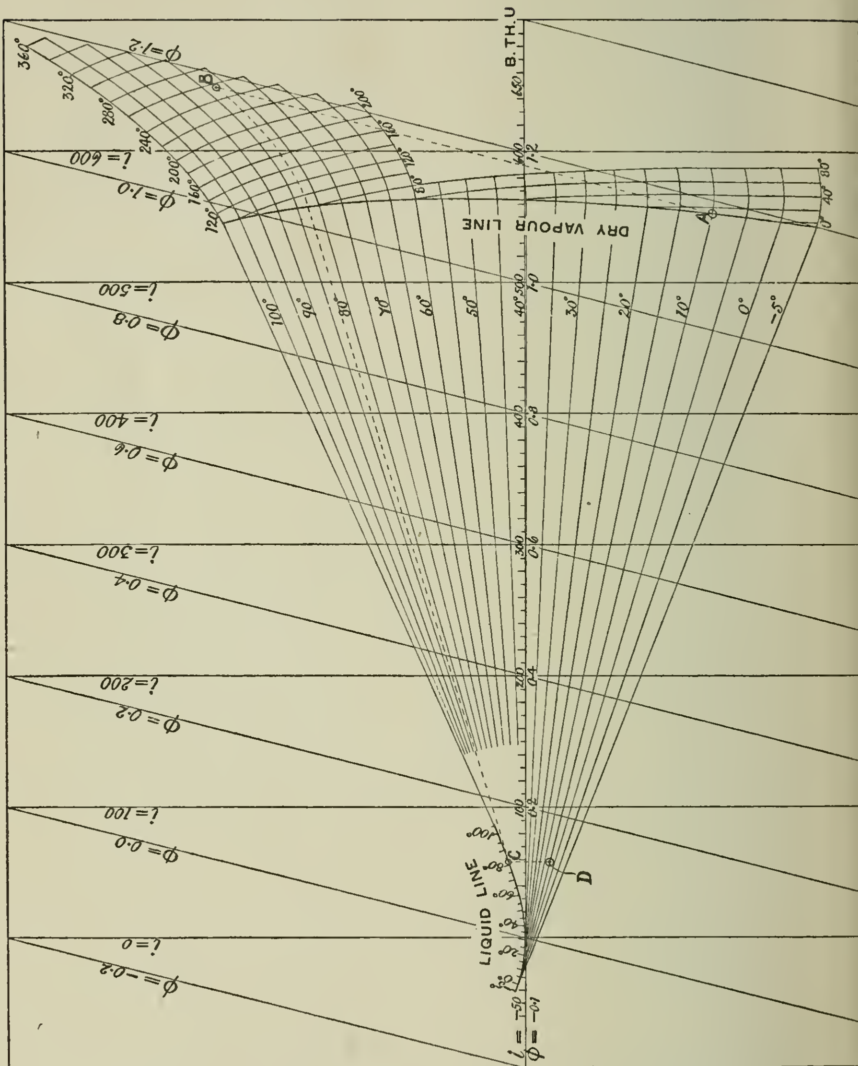



FIG. 42.—$\phi$-$i$ Diagram for $NH_3$.

[Note.—The values of $\phi$ and $i$ on this diagram for $NH_3$ were made with data not so recent or good as that given in the Table of the properties of $NH_3$, but these differences will not affect greatly any deductions made from this diagram.]

A similar cycle to the one just described on the $CO_2$ diagram, and having the same letters A B C and D to represent the same conditions of the refrigerant as shown would give the following values of $i_A$, $i_B$ and $i_C$, 552·8, 648·5 and 58·2 respectively, giving $W = 95 \cdot 7$ and $R = 494 \cdot 6$ B.Th.U.

The use of these diagrams has facilitated the author's calculations on the vapour-compression refrigerating machine.

The above described $\phi$-$i$ diagrams are of no use however for the suggested cycle with isothermal compression, and use has been made of a new diagram with $i$ as one co-ordinate and a thermodynamic function $\psi$ as the other co-ordinate. This function $\psi$ (to be defined later) has the property of representing the work done in the compressor in isothermal operations in precisely the same way that $i$ represents the work in the compressor in adiabatic operations. Ordinary rectangular co-ordinates suffice, and the Fig. 43 (page 1048) shows such a diagram for $CO_2$ and Fig. 44 (page 1049) for $NH_3$. Taking for example a $CO_2$ machine working with the upper pressure 1,000 lb. per sq. inch, corresponding vapour temperature 82·5° F., isothermal compression, and the transfer of heat from the relatively hot liquid before free expansion to the gas about to be compressed, effected in the manner described for the proposed cycle, the operations would be represented as follows. Let A represent the dry vapour which will receive heat from the hot liquid to raise its temperature to practically 82·5° F. at constant pressure. B will then represent its conditions. Isothermal compression brings it to condition represented by C, and condensation brings it to condition D at the same pressure and temperature. The liquid is then further cooled before free expansion in the manner described, but it is unnecessary to follow the operation to find the drop in temperature, for if $i_B$ and $i_D$ represent the values of $i$ in the refrigerant in conditions B and D, and $\psi_B$ and $\psi_C$ are values of $\psi$ at B and C, then the refrigerating effect $R = i_B - i_D$

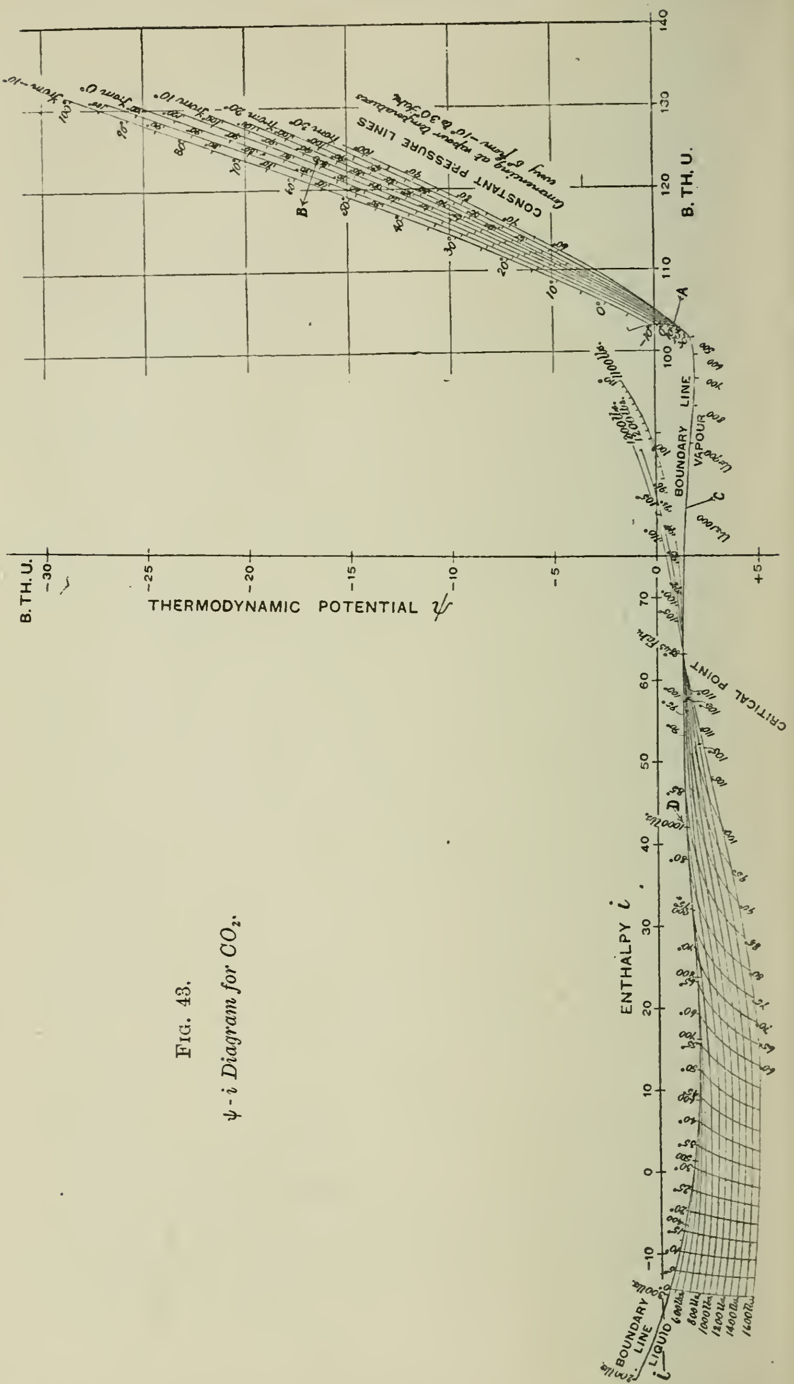



FIG. 43.

*ψ-i Diagram for $CO_2$.*

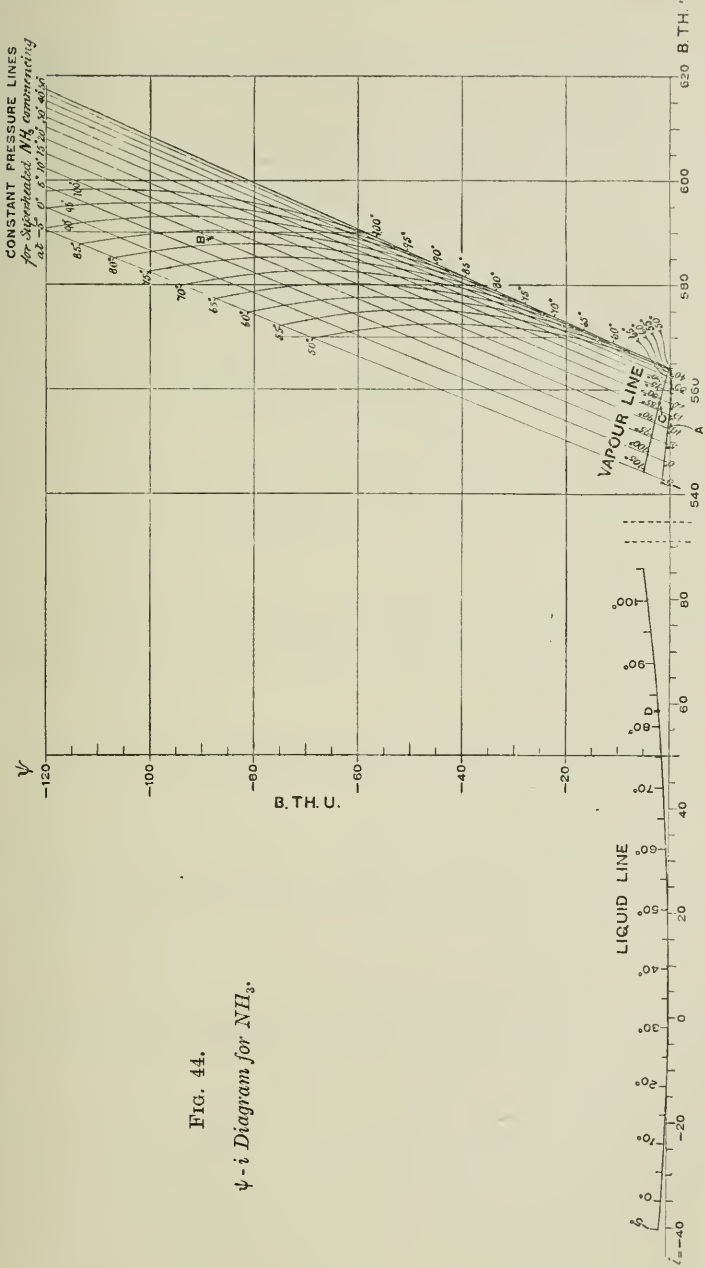



Fig. 44.

*$\psi$-$i$ Diagram for $NH_3$.*

$CO_2$.

| $t$. | $p$. | $\frac{dp}{dt}$. | $v$. | $s$. | S. | L. | $Aps_o$. | $\phi_w$. | $\phi_s$. |
|---|---|---|---|---|---|---|---|---|---|
| ° F. | Lb. per sq. in. | Lb. per sq. in. | Cubic ft. | Cubic ft. | | | | | |
| −5 | 285 | 4·702 | 0·317 | 0·0160 | −17·60 | 119·87 | 0·934 | −0·0371 | 0·2637 |
| 0 | 310 | 5·011 | 0·291 | 0·0161 | −15·40 | 117·62 | 1·014 | −0·0323 | 0·2559 |
| 5 | 335 | 5·334 | 0·266 | 0·0163 | −13·16 | 115·25 | 1·098 | −0·0275 | 0·2481 |
| 10 | 363 | 5·671 | 0·244 | 0·0165 | −10·87 | 112·75 | 1·188 | −0·0226 | 0·2401 |
| 15 | 392 | 6·024 | 0·224 | 0·0167 | − 8·52 | 110·10 | 1·284 | −0·0176 | 0·2320 |
| 20 | 423 | 6·392 | 0·205 | 0·0169 | − 6·11 | 107·29 | 1·386 | −0·0125 | 0·2237 |
| 25 | 456 | 6·776 | 0·188 | 0·0172 | − 3·62 | 104·31 | 1·494 | −0·0074 | 0·2153 |
| 30 | 491 | 7·177 | 0·172 | 0·0174 | − 1·05 | 101·14 | 1·608 | −0·0021 | 0·2066 |
| 35 | 528 | 7·593 | 0·158 | 0·0177 | 1·61 | 97·75 | 1·729 | 0·0033 | 0·1976 |
| 40 | 567 | 8·027 | 0·144 | 0·0180 | 4·37 | 94·11 | 1·857 | 0·0088 | 0·1884 |
| 45 | 608 | 8·479 | 0·132 | 0·0183 | 7·25 | 90·19 | 1·992 | 0·0146 | 0·1787 |
| 50 | 652 | 8·948 | 0·120 | 0·0187 | 10·28 | 85·93 | 2·134 | 0·0206 | 0·1686 |
| 55 | 698 | 9·436 | 0·109 | 0·0192 | 13·49 | 81·27 | 2·285 | 0·0268 | 0·1579 |
| 60 | 746 | 9·943 | 0·099 | 0·0198 | 16·93 | 76·12 | 2·443 | 0·0335 | 0·1465 |
| 65 | 797 | 10·468 | 0·089 | 0·0205 | 20·66 | 70·34 | 2·610 | 0·0406 | 0·1341 |
| 70 | 851 | 11·014 | 0·080 | 0·0213 | 24·78 | 63·70 | 2·786 | 0·0484 | 0·1203 |
| 75 | 907 | 11·579 | 0·071 | 0·0222 | 29·49 | 55·82 | 2·971 | 0·0573 | 0·1044 |
| 80 | 967 | 12·165 | 0·061 | 0·0235 | 35·20 | 45·87 | 3·166 | 0·0679 | 0·0850 |
| 85 | 1029 | 12·771 | 0·050 | 0·0259 | 43·18 | 31·28 | 3·370 | 0·0826 | 0·0574 |
| 86 | 1042 | 12·895 | 0·047 | 0·0268 | 45·44 | 27·00 | 3·412 | 0·0868 | 0·0495 |
| 87 | 1055 | 12·990 | 0·044 | 0·0282 | 48·31 | 21·56 | 3·454 | 0·0920 | 0·0394 |
| 88 | 1068 | 13·146 | 0·040 | 0·0308 | 53·17 | 12·84 | 3·497 | 0·1009 | 0·0234 |
| 88·43 | 1074 | 13·200 | 0·035 | 0·0346 | 59·23 | 0·00 | 3·515 | 0·1120 | 0·0000 |

$NH_3$.

| $t$. | $\tau$. | Pressure. | $\frac{dp}{dt}$. | $v$. | S. | L. | H. | $\phi_w$. | $\phi_s$. |
|---|---|---|---|---|---|---|---|---|---|
| ° F. | Absolute. | Lb. per sq. in. | Lb. per sq. in. | | | | | | |
| −27·4 | — | 14·7 | — | — | — | 617·5 | — | — | — |
| −10 | 449·6 | 23·34 | 0·5888 | 12·20 | −45·94 | 599·6 | 553·7 | −0·0977 | 1·334 |
| − 5 | 454·6 | 26·44 | 0·6537 | 10·77 | −40·57 | 594·4 | 553·8 | −0·0858 | 1·307 |
| 0 | 459·6 | 29·88 | 0·7240 | 9·54 | −35·18 | 589·2 | 554·0 | −0·0740 | 1·282 |
| 5 | 464·6 | 33·69 | 0·7998 | 8·47 | −29·76 | 583·9 | 554·1 | −0·0623 | 1·257 |
| 10 | 469·6 | 37·89 | 0·8812 | 7·54 | −24·32 | 578·5 | 554·2 | −0·0506 | 1·232 |
| 15 | 474·6 | 42·52 | 0·9686 | 6·73 | −18·84 | 573 0 | 554·2 | −0·0390 | 1·207 |
| 20 | 479·6 | 47·60 | 1·0619 | 6·01 | −13·33 | 567·5 | 554·2 | −0·0275 | 1·183 |
| 25 | 484·6 | 53·15 | 1·1610 | 5·39 | − 7·80 | 562·0 | 554·2 | −0·0161 | 1·160 |
| 30 | 489·6 | 59·21 | 1·2661 | 4·85 | − 2·23 | 556·4 | 554 2 | −0·0046 | 1·136 |
| 35 | 494·6 | 65·81 | 1·3774 | 4·47 | + 3·36 | 550·7 | 554·1 | 0·0068 | 1·113 |
| 40 | 499·6 | 72·98 | 1·4948 | 3·95 | 8·98 | 545·0 | 554·0 | 0·0181 | 1·091 |
| 45 | 504·6 | 80·76 | 1·6184 | 3·57 | 14·63 | 539·2 | 553·8 | 0·0293 | 1·069 |
| 50 | 509·6 | 89·18 | 1·7481 | 3·24 | 20·31 | 533·3 | 553·6 | 0·0405 | 1·047 |
| 55 | 514·6 | 98·27 | 1·8839 | 2·95 | 26·02 | 527·4 | 553·4 | 0·0517 | 1·025 |
| 60 | 519·6 | 108·05 | 2·0253 | 2·69 | 31·76 | 521·4 | 553·2 | 0·0628 | 1·003 |
| 65 | 524·6 | 118·54 | 2·1721 | 2·46 | 37·52 | 515·3 | 552 8 | 0·0738 | 0·9823 |
| 70 | 529·6 | 129·77 | 2·3239 | 2·25 | 43·32 | 509·1 | 552·4 | 0·0848 | 0·9611 |
| 75 | 534·6 | 141·77 | 2·4806 | 2·065 | 49·14 | 502·8 | 551·9 | 0·0958 | 0·9405 |
| 80 | 539·6 | 154·56 | 2·6418 | 1·899 | 54·99 | 496·5 | 551·5 | 0·1067 | 0·9201 |
| 85 | 544·6 | 168·16 | 2·8072 | 1·750 | 60·88 | 490·1 | 551·0 | 0·1175 | 0·8999 |
| 90 | 549·6 | 182·60 | 2·9763 | 1·616 | 66·79 | 483·6 | 550·4 | 0·1283 | 0·8799 |
| 95 | 554·6 | 197·91 | 3·1485 | 1·496 | 72·73 | 477·0 | 549·7 | 0·1391 | 0·8601 |
| 100 | 559·6 | 214·11 | 3·3231 | 1·387 | 78·70 | 470·3 | 549·0 | 0·1498 | 0·8404 |
| 105 | 564·6 | 231·23 | 3·4993 | 1·289 | 84·69 | 463·4 | 548·1 | 0·1605 | 0·8208 |

and the work done in the compressor $W = \psi_B - \psi_C$ for the reasons previously given. The actual figures are $i_B = 123 \cdot 8$, $i_D = 42 \cdot 1$, $\psi_B = 16 \cdot 45$ and $\psi_C = -1 \cdot 42$, so that $R = 81 \cdot 7$ and $W = 17 \cdot 87$, and $\eta = 4 \cdot 58$, an increase of 40 per cent. for the new cycle.

A similar cycle on the $\psi$-$i$ diagram for $NH_3$ would be represented by A B C D. . ., the same letters denoting similar conditions on the two diagrams.

*Proofs of the conclusions named when referring to the functions $i$ and $\psi$, the former being called the enthalpy.*—Commencing with the fundamental thermodynamic relation, using heat units throughout,

$$dH = dU + pdv \quad . \quad . \quad . \quad . \quad (1)$$

or
$$dH = \tau d\phi = d\,(U + pv) - vdp$$

write
$$i = U + pv,$$

so that
$$dH = di - vdp \quad . \quad . \quad . \quad . \quad (2)$$

In an adiabatic operation $dH = O$, so that $di = vdp$, and between limits

$$i_2 - i_1 = \int_1^2 vdp,$$

and the right hand side of this represents at once the area of the work-done diagram in the compressor, so that the change of $i$ measures the work done in the compressor.

From the equation (2) if $dp = O$, that is, during a constant pressure operation, $di = dH$, so that the second property of $i$, namely, that it measures the heat taken in or rejected during constant-pressure operations, is proved.

That $i$ is unchanged by a free expansion operation is obvious from its definition, for $i = U + pv$ by definition, and since $U + pv$ measures the whole stock of energy possessed by the substance, and no work being done or heat supplied to the substance on the whole during a free expansion operation, the value of $U + pv$ will be unchanged in the substance passing through the orifice, and hence $i$ remains unchanged.

Now consider the thermodynamic function $\psi$ defined by the equation

$$\psi = i - \phi\tau$$

thus since from (2)

$$\tau d\phi = di - vdp,$$

or

$$d(\tau\phi) - \phi d\tau = di - vdp,$$

or

$$d(i - \tau\phi) = - \phi d\tau + vdp,$$

we get the result

$$d\psi = - \phi d\tau + vdp.$$

In an isothermal operation $d\tau = 0$, so that

$$d\psi = vdp,$$

and between limits

$$\psi_2 - \psi_1 = \int_1^2 vdp,$$

so that changes of $\psi$ represent the work done in the compressor if the compression is isothermal.

The Paper is illustrated by 7 Figs. in the letterpress.

[*The Discussion on this Paper was combined with that on the Paper by* Mr. J. Wemyss Anderson, *and commences overleaf.*]

*Discussion on Friday, 22nd November* 1912.

The President proposed a hearty vote of thanks to the authors for their very interesting, instructive and useful Papers, which he hoped would lead in the future to the clearing up of the various difficulties which the subject presented.

The vote of thanks was carried with acclamation.

Professor C. Frewen Jenkin (Oxford University) thought all would agree with him in saying that Mr. Anderson's Paper collected together a very useful summary of the present practice in refrigeration in all its subdivisions, and he also thought all would agree that Dr. Grindley's contribution indicated that he had done a great deal of work in the preparation of Tables such as had been placed before the Institution and that his new diagrams were very ingenious and interesting. He (Professor Jenkin) desired, however, to offer some perhaps rather severe criticisms on both Papers, but he hoped the authors would not think he did not value them on that account.

He thought it was a great pity that both authors had used the Fahrenheit degree instead of the Centigrade degree. He was not an advocate of the Continental units; he thought they were usually quite as troublesome as British units, which were bad enough, but he did consider that the Centigrade degree had an enormous advantage. Without adopting the kilogram or the metre or any of the foreign units at all, the use of the Centigrade degree at once made all the Continental diagrams similar to the English ones. The Continental diagrams were given for calories per kg. of stuff, that was, kilogram-centigrade degrees per kilogram; and the English diagrams were given for lb.-degrees per lb. The result was the two weights cancelled out and the diagrams were identical. Also the latent heat, the total heat I, the entropy and the specific heats were the same in English, French, and German if the Centigrade degree was used. The Fahrenheit units were now being omitted from most of the books. Ewing's "Steam Engine"

and "Mechanical Production of Cold," and the Cambridge Steam Tables, and Wimperis's books amongst others, were all in Centigrade degrees, and he thought it was a step backwards to use the Fahrenheit degree.

The Paper he had most to say about was Dr. Grindley's. He thought Dr. Grindley's ingenious suggestion was really almost impossible, and he was very sorry that the author had not given any hint as to how it could be carried out. Dr. Grindley suggested that isothermal cooling would give an enormous advantage. His interchanger of heat was of no use at all unless at the same time one could do isothermal cooling. All were familiar with the great advantages of isothermal cooling in air-compressors. Most makers used duplex or triplex compressors, compressing in stages and cooling between as an approximation to isothermal compression, but he had not heard of anybody who had succeeded in compressing air isothermally, and it seemed to him to be impossible. He had prepared five diagrams to show the differences in the cycles, and what could be done and what could not be done.

Fig. 45.

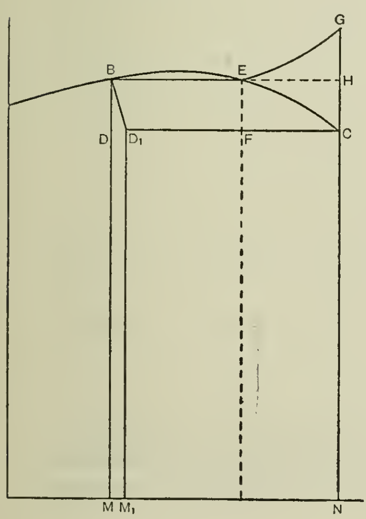
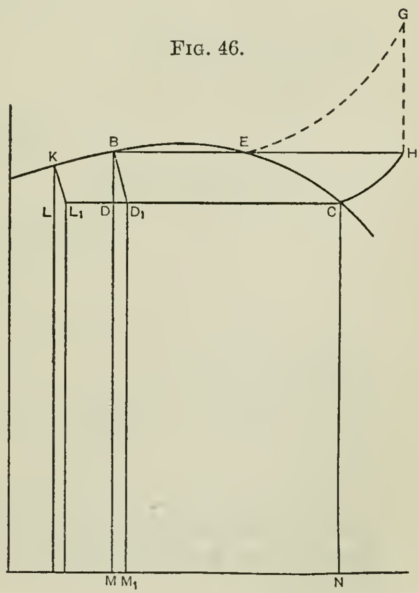



Fig. 46.



(Professor C. Frewen Jenkin.)

Fig. 45 was drawn to the same scale as Dr. Grindley's and showed the three ordinary cycles, and corresponded to the third example in his Table (page 1037).

B $D_1$ F E was the wet compression cycle.

B $D_1$ C G E „ dry „ „

B $D_1$ C H E was a special cycle, assuming that isothermal compression was possible; the isothermal compression was represented by the line H E; it could not begin at C because the cooling water was not cold enough. This cycle saved the work represented by the triangle H G E.

Fig. 46 showed Dr. Grindley's proposed cycle, K $L_1$ C H E; the line C H showing the effect of the interchanger.

Finally, K $L_1$ C H G E showed what would happen if isothermal compression was, as he believed it was, impossible.

The results of these five cycles were shown below:—

| | $\frac{R}{W}$ | Heat rejected. | Relative Volumes. |
|---|---|---|---|
| Cycle 1. Wet compression . . . | 5·09 | 0 | 15 |
| 2. Dry „ . . . | 5·05 | 0 | 19 |
| 3. „ „ & isothermal in part | 5·45 | 13·7 | 19 |
| 4. Dr. Grindley's Cycle . . | 6·2 | 24·8 | 21 |
| 5. „ „ „ adiabatic compression | 5·04 | 0 | 21 |

The Table* showed that the interchanger did no good without isothermal compression.

The second column showed the heat rejected in the cylinder—if such rejection were possible.

* The comparison of the volumes of the cylinders made in the foregoing Table was not quite fair to Dr. Grindley. The cylinder volumes should be compared "per unit refrigeration." On this basis Dr. Grindley's cycle would be the best.

The third column showed the relative volumes of the cylinders per pound of refrigerant.

The larger size of Dr. Grindley's compressor was a serious disadvantage. He would be glad if Dr. Grindley would make some suggestion as to how isothermal compression could be carried out; it would be of great value not only in connection with refrigerating machines, but in connection with many other types of machine.

Turning to Mr. Anderson's Paper, he noticed on page 1000 the author did not include Question *g*; he only spoke of it, but he (Professor Jenkin) thought it was a most important matter. It referred to the need of accurate data for the refrigerants; it appeared to him of very great importance that accurate data should be obtained. He had himself been working for more than a year on $CO_2$, and had extended the range of the data available from $-30°$ C. down to $-50°$ C., and he believed he had corrected a good many small errors and had obtained more accurate figures than were given by Dr. Mollier, but of course he could not place them before the members that night.

Mr. Gardner T. Voorhees (Member, American Society of Mechanical Engineers) said he considered it an honour to be invited to speak on the two Papers that had just been read. He had very little criticism to make on Mr. Anderson's Paper, except that he might say a word or two in regard to American practice. In the first place, in America an absorption machine would in every case hold its own with a compression machine—at least, as the latter was at present constructed. The question of the relative values of wet and dry compression was dealt with rather vaguely in the Paper, but as that matter was to come up at a meeting of the Cold Storage Association in December he would not touch on it at that moment. In the United States it was not considered wise to superheat the vapour above the temperature of the refrigerator; if one did so the capacity fell off. It was usually found that if as much heat as possible was taken out from the vapour, the compressor did not do as much work as it otherwise would do.

(Mr. Gardner T. Voorhees.)

He wished to refer to a matter which he had noticed in all works and Papers on refrigerators. On page 1021, Mr. Anderson, in calculating the capacity of the compressor, came down from 550 to, say, 450 B.Th.U. net under certain conditions which were referred to in the Paper, one of the most important of which was the amount of vapour formed as the liquid passed the expansion-valve. But one very important item that was not mentioned in the Paper was the converse of what happened in the steam-engine. It was well known that, in operating a steam-engine, the steam consumption which should be obtained according to the indicator-card was not obtained. There was a great deal of loss due to the cylinder condensation, and that cylinder condensation was, at any rate with saturated steam, a direct function of the difference between the temperature of inlet and outlet steam, that is, the temperature of the steam coming in and the temperature of the steam of the exhaust. With the compressor just the reverse took place, and for exactly the same reason—that there was a certain amount of cylinder superheating which took place. The gas coming in from the refrigerator was superheated, and therefore was not so dense as it should be, did not weigh as much as it should, and it was also a function in the difference in the temperature of gas discharged and the temperature of gas sucked in to the compressor. That brought him to the question of volumetric efficiency. It was stated in the Paper that the one pound of $NH_3$ vapour, under suction pressure given, occupied 9·1 cubic feet. Actually, it was not 9·1 cubic feet, but a good deal more. In fact, for the particular conditions mentioned, the volumetric efficiency of the compressor was 85 per cent. and not 100 per cent. as stated. In other words the figures given, if it was presumed the other allowances should hold, would have to be increased in that ratio, or by $17\frac{1}{2}$ per cent.—quite a perceptible factor.

He had been greatly interested in Dr. Grindley's Paper. He himself had spent a considerable time in translating Mollier's metric figures into English figures and making a diagram, and he knew how many hours of tedious and hard work it meant. As he had pointed out in an article which was published recently in some of

the British papers, it was impossible for any refrigerating engineer to work out satisfactorily or analyse any problem on a $CO_2$ machine without the use of Mollier's diagram—a diagram which Dr. Grindley had translated into English units and put in his Paper. In addition, Dr. Grindley had published a very valuable ammonia diagram, and some extremely valuable new diagrams which must have required an enormous amount of labour. He had made a few notes on the new system that Dr. Grindley had proposed in the Paper for utilizing some of the cold or taking out some of the heat in the liquid coming from the condenser to the refrigerator, but Professor Jenkin had anticipated much of what he intended to say. Taking the average case—the same that Dr. Grindley took, 10° F. and an 82½° F. condenser for $CO_2$, as given on page 1037, the fifth line of the Table —it would be found (and although Dr. Grindley did not give the figure it could be arrived at by dividing $R_2$ by $R_1$) that 37 per cent. more refrigeration was obtained, and 43 per cent. more performance. That was put forward as showing the theoretical advantage of the new system. He (Mr. Voorhees) believed, however, that in practice it would be impossible, except with an unpractical slow-speed compression or an unpractical small bore and long stroke, to transmit the heat of compression through the cylinder wall to the jacket-water—to water of the same temperature as the condenser; and therefore in practice adiabatic compression instead of isothermal compression would result. In other words, he did not think isothermal compression could be obtained, but that the compression would follow the adiabatic line almost to the very last eighth of the stroke.

Referring to Mollier's diagram (page 1044), he had applied the following conditions to it. A practical exchanger would exchange heat from the liquid to the vapour, say, to within 12½° of the hottest liquid, which would give 70° vapour to the condenser. That would cool the liquid to 68°, and showed, by the same diagram, 28 per cent. more refrigeration; but also by the same diagram it required 35 per cent. more power, and obviously no gain resulted, but a loss. As, however, the vapour entered the compressor at 70° in place of 0°, and as the compressor discharged, for the same diagram, at

(Mr. Gardner T. Voorhees.)

222° in place of 155°, it was evident that in the first place the gas was less dense and therefore took up more volume for the same weight; and, secondly, there was a greater difference in temperature of the discharge gas from the compressor and the inlet gas to the compressor. Therefore the volumetric efficiency of the compressor would be less, and the compressor had to be larger. He estimated an 18 per cent. larger compressor would be necessary, which of course increased the power required in the same ratio, from 35 to 41 per cent.—that was, 41 per cent. more power than the old method. Comparing the refrigeration and power in the old and new methods, he found that the performance of the compressor would be 20 per cent. less; that was to say, thermal units extracted from the refrigerator, divided by the thermal units exerted in the compressor in the $CO_2$ machine, would, he believed (taking the figures in the fairest way and taking figures which he had used and on which he had predicted results which came out almost substantially as he had predicted) come out 20 per cent. less in the new system. That was not due to any defect in the theory. The theory was all right so far as it went, but it did not take into account the volumetric efficiency and the increase in bulk of the gas, and so forth. As the $NH_3$ compressor should be theoretically less helped by the new process than the $CO_2$ compressor, it seemed that in practice a loss in the performance of $NH_3$ compressors would also always result from the new process. A further reason why a loss would result, in addition to what he had stated, was that one could make an exchanger where the vapour from the refrigerator did not pass through a pipe surrounding the liquid pipe; it could be passed round the pipe in such a way as not to have so much friction; yet there would be a tendency for friction to occur between the evaporator and the compressor, which did not at present exist, which would still further reduce the suction pressure, and therefore reduce still further the capacity of the compressor below the figures stated.

Mr. F. A. Willcox (Dartford) said that his firm (Messrs. J. and E. Hall, Ltd.) regretted that they were unable to

supply Mr. Anderson with information as to the machines they manufactured and their practice in refrigeration, owing to great pressure of work at the time.

Referring to the Paper, on page 950 the following passage was found: "The 'enormous progress' has, however, been more of a commercial nature than a scientific one; indeed, no branch of mechanical science has received less aid in this country from research or from published accounts of practical progress than that of mechanical refrigeration." This was followed by a statement of the fields of possible research. These remarks of the author were astonishing, and he (Mr. Willcox) found it difficult to imagine on what evidence they were founded. It seemed to him that such a statement could only be the result of a lack of knowledge of what was going on in the large establishments, which devoted their energies to the manufacture of installations for the purpose of mechanical refrigeration. It would be of interest if Mr. Anderson would explain how he differentiated between progress of a commercial nature and progress of a scientific nature. Any development which resulted in the application of natural laws to the service of man was a scientific development, and it was none the less scientific because it happened to be of commercial value. Did the author mean to contend that the labour and thought involved in developing the multifarious applications of refrigerating machinery in the industries of the country were not of a scientific nature because the applications were of benefit to commerce, and would he ask the members of the Institution to believe that the development of the use of refrigerating machinery for the transport of perishable produce over long distances had been achieved without the aid of scientific knowledge and without the use of scientific methods? If so, he was afraid he would have great difficulty in persuading a body of practical engineers to accept his view. These great developments had been achieved by the combined efforts of the users, consulting engineers, and manufacturers engaged in the industry, and he had no hesitation in asserting that they were of a scientific nature.

(Mr. F. A. Willcox.)

Speaking of his own firm, whose energies had been principally devoted to the development of the $CO_2$ machine and its application for the purposes of refrigeration in marine and land practice, he would say that the machine, when they first came in contact with it, was of the laboratory type, and a very large amount of time, labour, expense, and scientific thought and investigation had to be expended upon it before it was converted into a practical machine. They did not rest there; the theoretical possibilities of the machine were thoroughly investigated, and investigations were undertaken to determine to what extent it was advisable to modify the machine with a view to improving the coefficient of performance, always keeping before them the two following points: (1) That a refrigerating machine must be absolutely reliable under all circumstances. The failure of a machine would involve such serious loss that any modification introduced for improving the coefficient of performance, which would adversely affect the reliability, would not be justified. (2) Since the machines had to work under conditions and in situations where they did not receive highly-skilled attention and where it was impossible to obtain immediate aid in case of breakdown, they must be simple in operation. Pursuing this object, they had carried out trials of the following nature:—

(1) Cooling the liquid as it left the condenser, by allowing it to run into a receiver and by evaporating part of it into an auxiliary compressor or one end of a compressor according to the system devised by Professor Windhausen. These trials were carried out in conjunction with Professor Windhausen. The same principle had recently been brought forward, in which the gas from the receiver was admitted into the main compressor by means of a mechanically operated valve or by using a compressor with circumferential ports at the end of the stroke.

(2) Compound compression with intercooling between the two stages.

(3) Compound compression with a suction interchanger, in which the liquid flowing from the condenser was cooled by the gas passing from the evaporator to the compressor.

(4) The ordinary machine fitted with an expansion cylinder.

(5) The ordinary machine working under all practical ranges of temperature.

As a result of their investigations, his firm continued to manufacture the machine in its simple form. He would point out that investigations on the use of the expansion cylinder, and he believed on other methods of liquid cooling, were carried out by Kramer of Augsberg, who, he thought, was connected with Messrs. Riedinger, and they still continued to manufacture machines on the same lines as Messrs. J. and E. Hall. Beyond these investigations they had recently carried out an extensive series of trials of ammonia-compression machines under all practical ranges of temperature. These trials were begun some weeks before the Papers by Messrs. Tegelmeyer and Shipley on "Wet and Dry Compression" were read at the last meeting of the Association Internationale du Froid, and the results obtained indicated quickly that there was some advantage to be obtained by working the machine with the discharge gas superheated, over the method then in general practice of maintaining the temperature of this gas at the outlet temperature of the circulating water.

Turning to the question of cold storage to which the author referred, about two years ago Messrs. J. and E. Hall put down at the Pathological Laboratory of the University of Cambridge an installation by means of which, in conjunction with Professor Sims-Woodhead, they had carried out a number of experiments on the preservation of chilled beef, paying special attention to bacteriological action and the effect of temperature and preservative agents. Further, in conjunction with some Australian friends, investigations had been made on defrosting frozen meat. At the present time in conjunction with Sir Charles Parsons they were making investigations of another character in connection with this industry.

He had referred to the work of his own firm because with regard to this he had positive evidence, but it was quite apparent, from the class of installation of other manufacturers and, he thought, common knowledge, that they had made investigations which had

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been of great value in the development of the industry. The author stated (page 999) that "British refrigerating machinery by the leading makers is quite in keeping with the best traditions of the British engineer, but it would be idle to contend that the machines and their method of working are not capable of improvement." Of course, English manufacturers would appreciate the acknowledgment that their engineering practice was up to the best British standard, and all of them would agree with the author that finality in the development of the machine had not been reached; but he (Mr. Willcox) most strongly contended that the scientific progress made by British manufacturers was as great as the progress made by manufacturers in any other country. On the Continent they found that such firms as Linde, Sulzer, Escher Wyss, Borsig, and many others had been working on these machines for many years and with the result that their practice was identical with that of British manufacturers. The same might be said of the United States of America, where probably more machines were constructed and in use than in any other country.

Mr. Anderson defined two outstanding wants in refrigeration: (1) A standard unit of refrigeration; (2) A standard machine of comparison. He understood that the object of satisfying these wants would be to benefit the manufacturer and user. In the speaker's opinion these wants were not experienced by either party, and any attempt to make these definitions would be of no practical value. It was his firm's experience that they seldom obtained two inquiries alike: one might be to make so much ice under one set of conditions, another to make it under different conditions, another to cool stores and make ice in addition, and so forth, and the machine and plant had to be separately estimated for each inquiry. It would be an improvement if the term "ton of refrigeration" were abolished and English speaking manufacturers defined the capacity of their machines in the same manner as was done by Continental manufacturers, that is, by stating the number of frigories extracted between given limits of temperature.

Referring to the items of research recommended (page 1000): (*a*) *The Relative Efficiencies of the principal Refrigerants at varying*

*Temperatures.*—Most of the large manufacturers now made both $CO_2$ and $NH_3$ machines and some made, in addition, $SO_2$ machines, and they were well aware of what each machine's capabilities were. It was admitted that there was a field for the application of each type of machine, and it was somewhat difficult to see what useful purpose would be served by this Institution expending its funds in this direction.

(*b*) *The Advantages and Disadvantages of both Wet and Dry Compression.*—Since Mr. Anderson had prepared his Paper, Tegelmeyer and Shipley had both read important Papers on this subject; and from the experience gained in his own firm's investigations, he would agree with Tegelmeyer that it was sometimes advantageous to work on one system, sometimes on the other.

(*c*) *The Value of Compound Compression.*—In this country the Linde British Refrigeration Co. and Messrs. Haslam had long made machines on this system, and presumably had quite satisfied themselves as to what the advantages were. On the Continent Messrs. Sulzer and other makers (he believed) had also made machines on this system which had been described in technical journals.

(*d*) *Charge of Gas.*—This was essentially a matter for practice, and he would say that all manufacturers knew what the effect was. Generally speaking, an increased charge or an overcharge within limits increased the output and the power required in the same proportion.

(*e*) *The Value of Cooling the Liquid Refrigerant by the Evaporator before Expansion.*—His firm had made trials on these lines and had found no practical advantage. He believed that trials had also been made in Germany, and as the system was never used, presumably their results had been the same. The system involved an expensive complication, and it could only be justified if it effected a substantial improvement in the performance of the machine, which was not his firm's experience.

(*f*) *Deterioration of Refrigerant.*—As regards $CO_2$, they had never had the question raised, and they had now supplied

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considerably more than 3,000 machines. As regards $NH_3$, he had heard it stated by one engineer having charge of several large plants in Australia that deterioration did take place, and it was asserted by one of their staff that the gas, if at high temperature, decomposed into hydrogen and nitrogen so that it could be ignited. Speaking from his own experience, after a machine had been running continuously for some three months, and for a considerable part of the time with condenser water over 90° F., he fixed a Davy lamp arrangement on a cock on a discharge pipe of the compressor, and while the gas was leaving the compressor at a temperature of 300° F. he applied a light with the result that the light was extinguished. This did not prove that deterioration did not take place, but the only way to make an investigation of this sort was on a fairly large plant in constant operation, and from which samples of gas and liquid could be taken at intervals and carefully analysed. The Paper was of undoubted interest, but it seemed to the speaker to be a pity that the author considered it necessary to make a remark which cast an unmerited reflection upon those in this country who had been concerned in the development of this great industry.

Mr. J. T. Milton desired to thank Mr. Anderson for having sent to him, as the President of the Cold Storage and Ice Association, an invitation to be present that evening. The invitation arrived while he was abroad, and as he had not been home very long, he had not been able to ascertain the views of the Cold Storage and Ice Association. But he felt certain that if the Institution of Mechanical Engineers did appoint a Committee of Research, and in that way carry on similar work to that which they had so usefully done in other branches of Engineering, the Cold Storage and Ice Association would be very pleased to nominate some one, if invited to do so, to represent the users.

Mr. R. J. Cracknell desired to refer to the criticism of Mr. Anderson with regard to the absorption machine. Mr.

Anderson said that that machine could not hope to hold its own against the compression machine. He (Mr. Cracknell) would give certain results that had been obtained with the absorption machine, which would show that that was by no means the case. The absorption machine would produce a ton of ice per day with 48 lb. of steam per hour, including the power for all auxiliaries. In an ice factory on the compression system the power required, including power for auxiliaries, was never less than 3 h.p. per ton of ice, and was sometimes as much as 4 h.p. Taking an ice factory, say from 20 to 50 tons ice-making capacity, and assuming a horse-power of 3½ per ton of ice (the total power was often 50 per cent. greater than the actual power required by the compressor), and assuming in such a case that the power was provided by a compound condenser engine using 16 lb. of steam per h.p., the steam consumption would be 56 lb. per ton of ice as against 48 in the case of the absorption machine. In a 20-ton ice factory recently erected by his firm (Messrs. Ransomes and Rapiers) in Holland on the absorption system, a test showed it was capable of making 15½ tons of ice per ton of coal. In that figure was included power for all the auxiliaries, and for driving a dynamo for electric light and for supplying power to an adjoining workshop. In the first six months' running of that plant, taking the total ice made and total coal used and allowing for stoppages on Sundays, the result showed 14 tons of ice per ton of coal. For that same plant several Continental manufacturers of compression machines quoted prices, and the highest guarantee was given by a leading German firm, namely, 13 to 1.

He might also mention two instances where absorption machines had been installed during the past year in ice factories, which had hitherto been on the compression system. The compressors were shut down and the engine run to drive the auxiliaries, the exhaust going to the generator of the absorption machine. In both cases a very large saving of coal had resulted. Those instances were, he thought, sufficient to show that the absorption machine could more than hold its own. Of course in cases where exhaust steam was available, as suggested by the author of the Paper, no question of

(Mr. R. J. Cracknell.)
competition would arise, as with the absorption machine, results could be obtained which were not possible in any other way.

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*Discussion on Friday, 20th December 1912.*

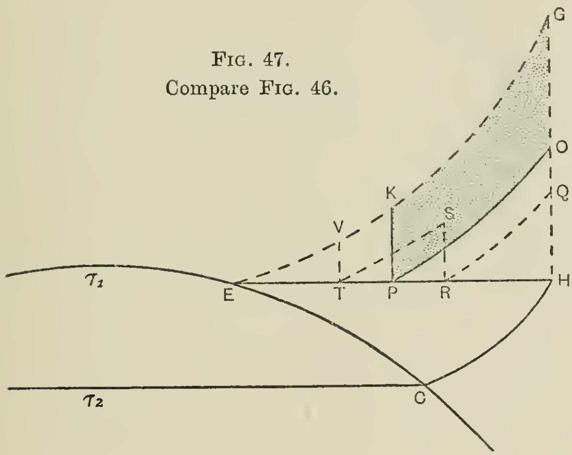
Captain H. Riall Sankey (Member of Council) desired to call attention to Appendix II (page 1003) of Mr. Anderson's Paper, where the principles of refrigeration were set out in comparatively simple differential equations. He looked upon this Appendix as a most excellent example of the use of combined analytical and graphical methods, and he thought the author was to be congratulated on the way in which he had treated that part of the subject. The differential equations could be solved, but then they became exceedingly long and cumbrous expressions which very few could readily follow. But when they were put in a graphical form quite a simple Figure was obtained, such, for instance, as Fig. 35 (page 1013). The curved and straight lines in that Figure were the graphical analogues of the long and cumbrous expressions mentioned above.

As an example of the simplicity of graphical solutions, he would refer to Fig. 46 (page 1055), drawn by Professor Jenkin, which was partially reproduced in Fig. 47. The curve represented the $\theta\,\phi$ chart for $CO_2$ ; $\tau_1$ was the higher temperature and $\tau_2$ was the lower temperature. Dr. Grindley was desirous of compressing isothermally, that is, from H to E, in order to improve the "coefficient of performance" of the cycle. But Professor Jenkin pointed out that in practice the compression would probably be adiabatic, that is, along H G, requiring a large amount of work, so that the cycle would practically not be so efficient as the one now in use. If, however, the compression were done in two stages with intercooling, there would be adiabatic compression up to the point O; then cooling at constant pressure along O P, followed by adiabatic compression along P K, and cooling at constant pressure along K E, and the work represented by the area K G O P would be saved. If the

compression were carried out in three stages with intercooling, then a diagram like H Q R S T V E was obtained, showing a further saving in work; and obviously, with an infinite number of stages, isothermal compression along H E was obtained.

With reference to the rating of refrigerating machines, it seemed to him that the difficulty pointed out by Mr. Anderson was the same as that which occurred in connection with steam-engines when the performance was compared by the number of pounds of steam required for the feed. In the case of the steam-engine, that was not a true comparison, because the pounds of

Fig. 47.
Compare Fig. 46.




steam did not necessarily contain the same amount of heat, and that was particularly true if superheated steam were used. The Committee of the Institution of Civil Engineers on Engine and Boiler Trials adopted a term called the "equivalent feed"; a factor was obtained depending on the steam conditions by which the actual feed was to be multiplied, and thus the equivalent feed was obtained. This factor was the ratio of the heat supply to the steam, reckoned from the exhaust temperature divided by 1,100. For example, if the actual feed was 8·9 lb. with superheated steam, the factor might be 1·2, and hence the "equivalent feed"

(Capt. H. Riall Sankey.)
would be $8 \cdot 9 \times 1 \cdot 2 = 10 \cdot 68$: and that, compared with a saturated steam-engine using $12\frac{1}{2}$, would be the real comparison and not the $8 \cdot 9$ as compared with $12 \cdot 5$.

As regards the standard of comparison, it seemed to him that two criteria were wanted; the first, being for the user who desired to know what the cost of running his machine would be when cooling a certain amount of liquid, and that cost must include all losses due to subsidiary machinery, leakages, conduction, and so forth. That measure corresponded to the number of pounds of coal burnt per "kw.-hour sold," a general expression used in the case of a central electric station. But the engineer also wanted a standard of comparison, and what was that standard to be? It was generally taken as the Carnot cycle, or rather the reversed Carnot cycle; but Mr. Anderson had shown (page 1016) that the true cycle for a compression machine was the inverted Rankine cycle, and at the bottom of the page he gave the coefficiency of performance as $8 \cdot 2$. The Carnot cycle between the same temperatures, namely, 528° F. absolute and 474° F., had a coefficiency of performance of $8 \cdot 8$, or approximately the Rankine cycle was about 7 per cent. less efficient than the Carnot. On page 1018 the coefficient of performance for dry compression, that is, with superheat, was shown to be $7 \cdot 12$, but the Carnot cycle for the range of temperature between A E and B C on Fig. 37 (page 1017) gave a coefficient of performance of $8 \cdot 8$ as before, the temperature being the same. If the extreme ranges of temperature were taken, that is, the superheated temperature, which was worked out by the author as 682° F., then the coefficient of performance became $2 \cdot 28$. The coefficients of performance depended, therefore, very greatly on the ranges of temperature that were taken. Which of them was to be selected for a standard?

In his opinion the requirements for a standard were: (1) that the cycle of operations should be as nearly as might be that of the actual machine, having reference to the substance in use; (2) the standard should be free from all losses; (3) the data for computing the standard should be easily ascertained by simple measurement; (4) the expression for the coefficient of performance

must be a simple one. The first condition and the third and fourth were antagonistic; the standard would, therefore, have to be a compromise, and it seemed to him that the only satisfactory way would be to entrust the matter to a committee.

In conclusion, he would ask Mr. Willcox, who spoke on the last occasion, whether he could not give some information as to the results obtained in the extended experimental work that had been carried out by his firm at Dartford. The object of the discussions at the Institution was to obtain information from the members and from visitors. A mere recital of experimental work done, without giving any information as to results obtained, did not help very much.

Mr. G. T. Harrap thought the two Papers were of a most interesting character and dealt with a subject which had not been discussed for a considerable length of time. Dr. Grindley's contribution to the theory of Refrigerating Machines was a valuable exposition of the question, but he looked at it from his own point of view. During the sixteen years he (Mr. Harrap) was editor of *Ice and Cold Storage* there had been placed before him at times for publication a number of diagrams of a similar character to those shown in the Paper, and as he had to test them he quite appreciated the enormous amount of labour the author must have performed in producing such very excellent diagrams. The result, however, did not quite coincide with the ideas the author had in view, namely, that by precooling so much advantage was to be obtained. The idea of precooling was not new by any means; and, although on special occasions it was used at the present time, he knew of no invention brought out during the last thirty years in connection with the matter which was now in use. The effectiveness of applications was dependent upon the economy of the entire system. It was of no advantage to say that, from one particular portion of the system, economy could be obtained if it was lost in another. Hence, precooling was not used nowadays. At the same time it must be acknowledged that Dr. Grindley's efforts had produced some very valuable diagrams

(Mr. G. T. Harrap.)
that had not been available before, and there was no doubt that for educational purposes they would be of considerable value.

With regard to Mr. Anderson's Paper, he was equally struck with Mr. Willcox by the remarks made in the second paragraph at the top of page 950: "The 'enormous progress' has, however, been more of a commercial nature than a scientific one; indeed, no branch of mechanical science has received less aid in this country from research or from published accounts of practical progress than that of mechanical refrigeration." He himself had been associated with mechanical refrigeration for over thirty-five years, as a scientist originally, and that quotation did not agree with his view of what had happened. Since Harrison's work of 1855–1860, and his memorable patent of 1857, he did not know of any progress that had been made except that due to scientific research, either independently or by the actual manufacturers themselves. Whether they had cared to give away that information or not was another matter, but the fact still remained, as he knew from experience, that researches had been made by a number of manufacturers with the assistance of very good scientific experts.

With regard to the statement made by Mr. Anderson with reference to the absorption machine (page 950), he thought it was only fair to point out that that machine still possessed great possibilities, and it was being used economically at the present time with good results. There was no doubt about it from actual experience that it possessed many possibilities for the future. The actual calculations given in the diagrams seemed to run very much on the same lines as those dealt with for many years past, but he was very sorry to find that two of the principal manufacturers were not represented by diagrams and illustrations of their machinery—Messrs. J. and E. Hall and Messrs. Linde. Messrs. Hall's representative had already explained why his firm did not send the diagrams, but apparently there was nobody present to explain why Messrs. Linde had not sent diagrams. He understood from Mr. Anderson now that they were very busy at the time and were not in a position to provide the information.

Passing over various detailed matters referred to in the Paper, which he did not think for present purposes it was necessary to refer to, it seemed to him the principal point was the conclusions arrived at, apart from the question of the discussion, as to the use of dry compression and wet compression, with reference to which he thought it was only necessary to say that the original machines were calculated and built on a dry-compression basis, and were ultimately worked on a wet-compression basis from purely practical considerations. He knew that one of the machines at least had been working for thirty years on a wet-compression basis, and it had never had a cylinder bored; it was still running as efficiently as ever, which proved that at any rate, in adopting the wet-compression basis, practice in that particular instance overrode theory. He did not for one moment wish to contend that the compressor as a compressor was more efficient with wet compression than with dry compression; but taking the whole of the circumstances into account he was prepared to state very strongly that wet compression over a term of years would show the best results. In that connection no better test could be obtained than in the manufacture of ice. If two machines, wet and dry, were tested over a series of, say, five years, he was certain from his own experience that, if the whole of the costs were taken into account, the wet compression would show the greater advantage.

He thought all present would agree that, from a scientific point of view, there was no doubt it would be a great advantage to have a determinate measure—a standard. In the two different standards which Mr. Anderson mentioned there was not sufficient difference to call for any very great discussion—he believed the difference amounted to 1 per cent. Bearing in mind that manufacturers had still managed to go on with the horse-power as a standard of measure, which he supposed was about 50 per cent. out, and bearing in mind also that the kilogram was supposed to be such a marvellous improvement, it made one think after all whether it was necessary to determine a standard with absolute accuracy, so long as a standard of some kind was agreed upon. Having done that, what was its application? Had anybody asked for it? Had the manufacturers

(Mr. G. T. Harrap.)

asked for it? Had the people who purchased the machines asked for it? He had never heard from either one or the other that there had been any demand for a standard, although he believed from a theoretical point of view it would be a very good thing to have one for educational purposes at any rate. But in fixing a standard it must not be forgotten, from the point of view of mechanical engineers, that the manufacture of the plant and the machinery was the least important part to be considered. Next in importance came the insulation, and the most important part of all was that of application; and the difference in application, based entirely on empirical considerations, was so great that the question of even 20 per cent. would not enter into the calculation. It had to be borne in mind that, while from an educational point of view there was no doubt it would be an advantage to be able to speak in set terms, when once a man entered into the commercial side they would be merely used as a kind of general reference; and unless the person using that reference had the knowledge obtained from a number of years' experience of how to apply his particular machine, he would not get the order. While it might be treated from the very highest theoretical point of view, it must not be forgotten that the touchstone of the whole thing was, how much money was the manufacturer going to get out of it? Therefore, while most highly appreciating the efforts of Mr. Anderson, who had taken considerable trouble in dealing with the position from an educational point of view, the question of application so overrode everything else that he was afraid it would be necessary to forget a very great deal of what one had previously learnt, although there could be no doubt that, from an educational point of view, some kind of standard on which to teach the young engineer should be obtained. He had found the same difficulty himself forty years ago when he started, and he was simply pointing out to those present what they might expect.

Professor Jenkin mentioned (page 1054) at the last Meeting that manufacturers ought to work on the Centigrade scale; but when he pointed out that by far the great majority—he would say 90 per cent.—of the people who purchased refrigerating machinery knew

nothing about Centigrade or Fahrenheit, the members would appreciate the fact that they had to deal with people as they met them. The manufacturers had to supply machines as they were required to do what the purchaser wanted, and the question of whether the compressor was of this or that type, or had such and such a rating, was after all a very small one.

Mr. A. H. Tyler said that, as a visitor, he was glad to have the opportunity of attending the Meeting and speaking on this subject. As previous speakers had remarked, both the Papers contained valuable information and dealt with matters that would bear a great deal more discussion than could be given at one Meeting. He hoped, as a result of the reading of the Paper and the discussion, that more publicity would be given to the question of the theory of the refrigerating machine, because it was much needed.

With regard to the question of publicity, two different opinions had been expressed. Mr. Anderson expressed his regret that so little experimental work had been done by the manufacturers; whereas one of the speakers (Mr. Willcox) at the last Meeting took up an antagonistic position and said that a great deal of work had been done by his firm. He thought both the author and the speaker were right. Unfortunately, however, the greater part of the results of the experiments that had been made by the manufacturers had not been made public, and a considerable amount of what had been published was quite useless. A good many of the experiments that had been made, the results of which had not been given to the public, were equally worthless. Many years ago some figures were published as to the properties of $CO_2$ that misled manufacturers considerably, which was very regrettable. Since then Mollier had experimented on the matter and had given manufacturers a sound basis to work upon, and he could not be thanked too much for what he had done in that respect. Following on Mollier's investigations, Amagat had carried the matter considerably further, and, having checked the majority of the figures obtained, so far as he could find they were perfectly accurate and reliable within ordinary limits. Gooseman had also

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published some very useful information about $CO_2$ that he had only seen referred to on one occasion in this country.

Since Mr. Anderson's Paper was written, he (Mr. Tyler) had had the opportunity of publishing a certain amount of information in another place, and probably members had seen it mentioned in the technical Press. Most of the information that had been published on $CO_2$ was so unreliable that personally he had to go back to the beginning and endeavour to establish some of the more elementary facts on a reliable basis. He hoped that manufacturers would join him in publishing results of tests; and he did not believe any good resulted either to the manufacturers or to anyone else by keeping results secret.

Mr. Harrap (page 1073) had referred to the question of units, stating that they were not required commercially, but he (Mr. Tyler) differed entirely from him as to their utility. The establishment of units was not proposed simply for educational purposes; they would also be of the utmost use to manufacturers. Almost every manufacturer had some standard or unit, but they nearly all differed. Personally, he suggested the adoption of a unit which in his opinion had much to recommend it, and he would be very pleased if gentlemen who differed from him would state the reason for doing so. The unit he had adopted for many years past for his own use as the Ton of Refrigeration was the elimination of 12,000 B.Th.U. per hour from brine at the temperature of 32° F. 12,000 B.Th.U. was almost exactly the American ton of 2,000 lb. multiplied by the latent heat of water, and divided by 24 hours per day. It assumed the latent heat of water at 144 B.Th.U. per lb. It was a trifle less, but that was the nearest even figure. The figure of 144 had advantages; it was easily divisible; it was a nice round figure, one easy to remember and use. Scientific accuracy in a standard was not of any importance whatever. It was like the horse-power; every one used it, but it did not represent the power of any standard horse that he knew of. Taking the figure of 32° F., as the temperature of the brine from which the heat was abstracted, had the recommendation that it compared with the thawing of ice or freezing of water, both of which took place at a temperature of

32°. It was very difficult to get water to freeze at any other temperature but 32° F., and if that was done, abstruse corrections had to be introduced. If a machine which was abstracting heat at zero was compared with a machine in which ice was changing at 32°, a correction was involved, and he was afraid that it would take the members a very long time to agree what proportion it should be.

Personally, he saw no objection to the adoption of 32° F., which also had another very great advantage. If the machines (either $CO_2$ or $NH_3$) were tested by abstracting heat from brine at 32° for shop test purposes, it would be found they took their maximum power with brine at about that temperature, with cooling water in the condenser up to from 70° to 80° F. So that by testing the machine with brine at 32° F., the maximum power was being used that would be required to be supplied by the motor working the machine under commercial conditions. As the machine had initially to cool brine from 32° downwards to its working temperature, the motor had to be made to do its duty with brine at that temperature. It was objected that most refrigerating duty was done below 32° F., but on the other hand there was a large number of refrigerating machines required for cooling water and for chilling work, and such refrigerating machines performed their work above 32° F. So that the ordinary duties of a refrigerating machine were sometimes below and sometimes above 32° F., and that was a fair average figure. As he had already stated, it needed no correction when compared with ice, and therefore, for standardizing his machines, he had adopted that standard of 12,000 B.Th.U. per hour abstracted from brine at 32° F., and he designed machines for ordinary climates to do that duty with condenser water up to 70° F. Cooling water of 70° F. was obtainable almost invariably in all temperate latitudes; it was a very usual summer temperature, and it was one that should be reckoned on. It was very seldom that water was not obtainable at that temperature, but in such special cases the difference could be allowed for. In the tropics it went very little above that figure as a rule.

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While on the question of units, he would like to suggest that some better terms should be adopted. The unit that was at present used was called a "ton of refrigeration," and he wished that some other term could be used in its place. Electricians had much more convenient words in the "volt," the "ampere" and the "ohm," the second of which was usually abbreviated into the "amp." Instead of the term "ton of refrigeration," he had adopted the words "frigid ton," which he thought was better than the other.

Mr. Anderson in the early part of his Paper referred to the cycle of performance of the refrigerating machine as being a heat-pump, and compared it with a reversed steam-engine. That was very usual, and he noticed that most writers on the subject treated it in the same way. He would like to suggest, however, with regard to Fig. 1 (page 952), that, when the operation of the machine was reversed, it was assumed that A remained the boiler and C the condenser; all that happened was that the boiler was at a lower temperature than the condenser. The evaporator was nothing more or less than a water-tube boiler, and the condenser was still a condenser, but at a higher temperature than the boiler. The result was that instead of giving out work in the cylinder it absorbed work; instead of absorbing work in the feed-pump it could give it out, and the pump was replaced by the expansion-valve. In the heat-engine the feed-pump must be a pump; the water could not be got to pass without it from the lower to the higher pressure; but with the refrigerating machine the pressures were reversed. An expansion cylinder, as fitted to the refrigerating machine, was an exact equivalent to the feed-pump reversed, and if it was looked at from the point of view he had mentioned, it would be found that a complete heat-engine reversed was obtained.

Some remarks had also been made on the question of wet and dry compression. The argument as to which of the two was the more advantageous had been raging for many years, and they seemed as far off settling the problem as ever. There was no doubt that some designs of $NH_3$ machines would give better results with dry compression. He was absolutely convinced from

the tests he had seen made that, under some conditions of working and design, ammonia machines gave better results with dry compression, but he could state most certainly, from experiments and tests he had made, that a $CO_2$ machine gave better results with wet compression than it did with dry. In one case he had had the opportunity of carrying out some tests in Alexandria, extending over about a month with a large $CO_2$ machine making ice. The water temperature was steady throughout the trial; the machine ran twenty-four hours a day and was only stopped once for some small adjustment. The ice was drawn in rows of moulds at intervals of about thirty-four minutes in the usual method followed in Continental practice. He tried the machine with varying degrees of wetness, until he found the most suitable degree. When he approached dry compression the results undoubtedly fell off, that is, he had to give a longer interval between drawing a row of moulds, showing that the machine was making less ice. The machine was indicated twice a day, and the results were very steady. The horse-power practically did not vary—it was within 1 per cent.—but the ice production certainly fell away as they approached dry compression. He had every reason to believe that, if they had gone on and got the compression completely dry, the results would have been much worse. With wet compression the results were better up to a certain point, and then they began to fall away again. He had found, from a very large number of tests of $CO_2$ machines, that the best results were obtained when a degree of wetness was used which gave a discharge temperature of about 150° F., and he thought that about 3° on either side of that figure covered the most economical temperatures under all conditions.

Dr. Grindley had explained the great increase in effect to be obtained from a $CO_2$ refrigerating machine, by cooling the liquid to the evaporation temperature in a "suction interchanger" and compressing isothermally, but he had unfortunately miscalculated the great increase of power consumed that would result. The entropy diagram dealt with weights of $CO_2$, not volumes, and the power required by the compressor was determined by the volume

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and pressures to be dealt with. Taking one example from his Table (page 1037), say the performance with temperature limits of 10° and 82·5°, the volume of the unit weight of gas (if dry) would be increased by 93 per cent. and the power for compression similarly, making $W_2 = 34 \cdot 5$ instead of $17 \cdot 9$ and $\eta_2 = 2 \cdot 38$ instead of $4 \cdot 57$; the increase in performance then disappeared and became a loss of nearly 26 per cent. With wet compression the loss would be greater.

In conclusion, he desired to thank the authors of the Papers for the work they had done and the results given, and although he had said nothing favourable about Dr. Grindley's Paper he thought it was extremely valuable. The amount of work involved in the diagrams that had been worked out was enormous, as he knew from having carried out similar work himself; and although he was a manufacturer and a practical man, he fully appreciated the importance of the theoretical side of the subject. He went as deeply into the one side as the other, and scientific gentlemen who gratuitously devoted their time to investigation and made known the results of their labours and prepared diagrams did the manufacturers an inestimable service.

Mr. G. C. Hodsdon said that since the Annual Report of the Institution for 1911 appeared he had been anxiously awaiting the present Papers on Refrigeration, and he was very surprised when they arrived to find that they treated the subject from the point of view of the ordinary engineer, for whom they were full of information. There was nothing for the refrigerating engineer. He presumed that the object of the Papers was that they should be of use to the refrigeration industry, and in order to accomplish that purpose they must contain something that practical refrigerating engineers could use in their work. Mention was made of research in connection with the subject, and he for one would be very pleased if the Institution could take up that work. There were, in his opinion, one or two subjects that could be dealt with most usefully. What were particularly required were figures that all refrigerating engineers could accept as standard.

Dr. Grindley referred to investigations that had been carried out, but did not say definitely what they were. Presumably they were connected with the new cycle which was proposed, and he would like Dr. Grindley to say whether that was so or not. The inference contained in the Paper was that refrigerating engineers did not know their business. That remained to be seen. First of all, Dr. Grindley put a water-jacket on to the compressor. If that was all he did, it was of no use; but perhaps the author had some other way of getting isothermal compression. His own experience was that isothermal compression could be obtained, but life was not long enough to wait for it, and it was not a commercial proposition. The machines had to be built, not for the express purpose of giving satisfaction to the refrigerating engineer, but to make money. Before vapour-compression machines became so common, cold-air machines were used, with the cycle of which most of those present were familiar. Air was taken at atmospheric pressure and temperature and compressed in a cylinder. It was then cooled whilst at a pressure of from 45 to 60 lb. per square inch in a tubular cooler, by water, to a temperature a few degrees above that of the temperature of the inlet cooling water. The compressed cooled air was then led to an expansion cylinder and make to do work, with the result that the temperature at the outlet of the expansion cylinder was somewhere about − 80° F. or even lower. In order to reduce as far as possible the power required for compression, the compression cylinder was always fitted with a water-jacket. In one particular case a thin gun-metal liner was fitted, the water circulating in the space left between the liner and the body. The size of the compressor was $11\frac{3}{4}$ inches diameter by 14 inches stroke, indicating 17 h.p. at 85 revolutions per minute, with an air-pressure of 45 lb. per square inch by the gauge, giving a ratio of compression of 4·06. A $CO_2$ machine or an ammonia machine, working between the limits given in Mr. Anderson's Paper of 0° F. and 70° F., would give a ratio of compression, for comparison, of 2·75 for the $CO_2$ machine and 4·28 for the ammonia machine. After everything had been done to keep down the temperature of the air discharged, it left the

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cylinder at 290° F., although it entered it at about 60° F. That was with a thin gun-metal liner. Was there any hope of isothermal compression for an ammonia machine or a $CO_2$ machine where it was necessary to have a liner much thicker than that?

He doubted whether many engineers present had ever seen a water-jacket on a $CO_2$ machine; one's natural conclusion was that such machines had not been made with water-jackets. That, however, was altogether incorrect. Fig. 48, Plate 46, showed a compound $CO_2$ compressor with a water-jacket; it was the first $CO_2$ machine ever built in this country, and it was made in the year 1888. When a water-jacket had been put on the machine, the next thing was to keep it there. Engineers were well acquainted with what happened to the water-jacket of an oil-engine or a motor-engine if there was a frost, and that happened to several of the water-jackets on the first $CO_2$ machines that were built. They would have to be very careful in their regulation of the machine or the water-jacket would come off, as some of the original ones did. He did not say that was the reason the water-jacket was given up, but it no doubt had something to do with it. It was easy enough to regulate the machine, but some day or other the man in charge of the machine would be a little careless, and the jacket would be split. Incidentally he saw in the December No. of *Ice and Refrigeration*—a journal published in America—that on some American dry-compression machines the water frequently left the jackets at a lower temperature than it entered. The water-jacket was intended to cool the cylinder and not to cool the water, but that was what happened in many cases if carelessness was shown in regard to the regulation of the machine. Then Dr. Grindley said that he would use an interchanger on his machine. In his opinion no gain would result, provided the temperature of the gas discharged from the compressor was fixed; and it was fixed, as Mr. Tyler had already stated, at about 150° F. or 160° F. for practical reasons, the principal one being lubrication. It was also found that the machine gave better duty under those conditions. Having a fixed size and speed of compressor, a fixed delivery temperature

and fixed evaporating and condensing temperatures, the quality, or dryness fraction, as it was usually called, of the gas entering the compressor was therefore fixed, and therefore the weight of gas dealt with in unit time by the compressor was also fixed. What happened to the medium between the regulator, where it was a liquid, and the point at which it entered the compressor did not affect the maximum duty to be obtained from it. The liquid had to first cool itself from the fixed condensing temperature by partial evaporation to the evaporating temperature, and then to evaporate, and whether this took place in the evaporator or partly in the evaporator and partly in an interchanger, it could only absorb a fixed number of B.Th.U. per lb. from an external source. They started with a liquid at a fixed temperature, and it came back to the compressor as a gas with a fixed dryness-fraction, temperature, and pressure. The addition of an interchanger gave no increase, in fact it caused a distinct loss, because they had got the transmission of heat from the surrounding atmosphere into the interchanger.

A full Table of the properties of $CO_2$ based on the latest data available was published in *Ice and Cold Storage* for November 1912. Details were given, not for every 5° or every 9°, but for every degree F.

Coming to the Ammonia Table, Dr. Grindley had worked on some figures which appeared in Zeuner's "Technical Thermodynamics," and he stated that they did not fit a smooth curve, which was not to be wondered at. A curve from the original figures given in 1890 by Zeuner would be seen in Fig. 49 (page 1084) marked Zeuner No. 1. In the year 1904 Dieterici published some figures, but they only went down to 32° F. Zeuner in his "Technical Thermodynamics" gave these figures (*see* the curve marked Zeuner No. 2 which only went down to 32° F.); he then continued with his older figures, thus causing a sudden jump to the old curve. He did not know whether the author had been able to test those curves of Zeuner's. The curve that Dr. Grindley had given was also shown and went outside the others at the bottom right hand corner, and he did not think that it was at all fair. The diagram gave ten curves. The various dates

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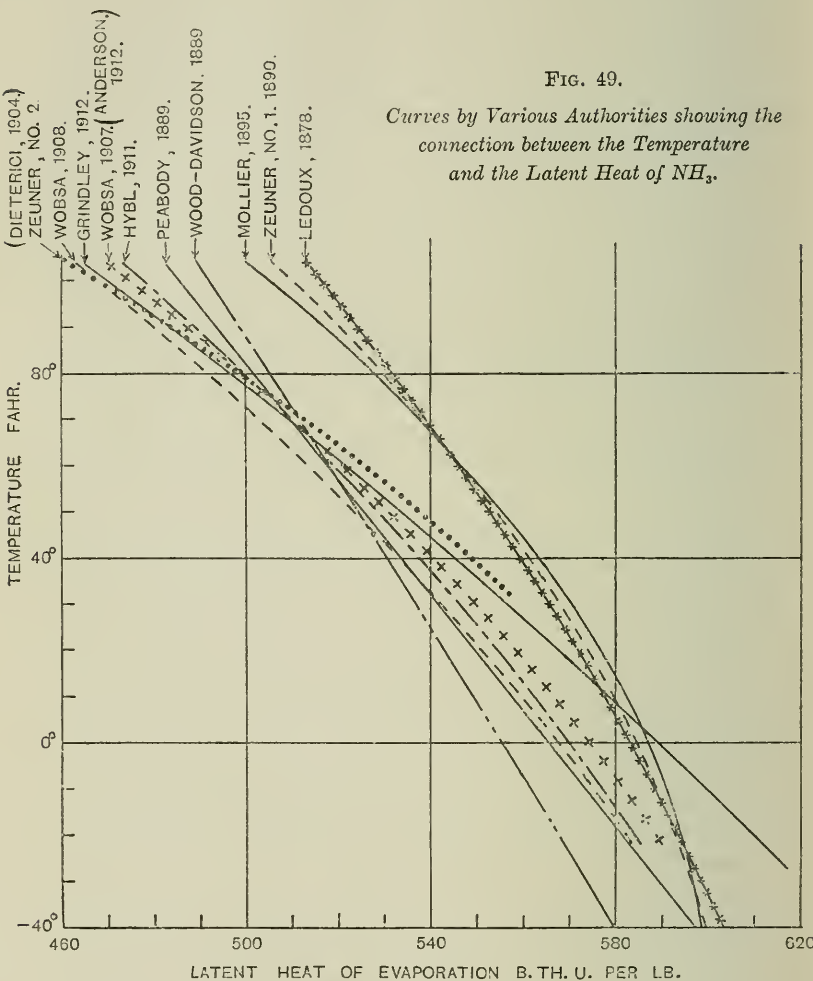
were given, and the authorities for them, and the members might take whichever they preferred. That was a point which the

Fig. 49.

*Curves by Various Authorities showing the connection between the Temperature and the Latent Heat of $NH_3$.*

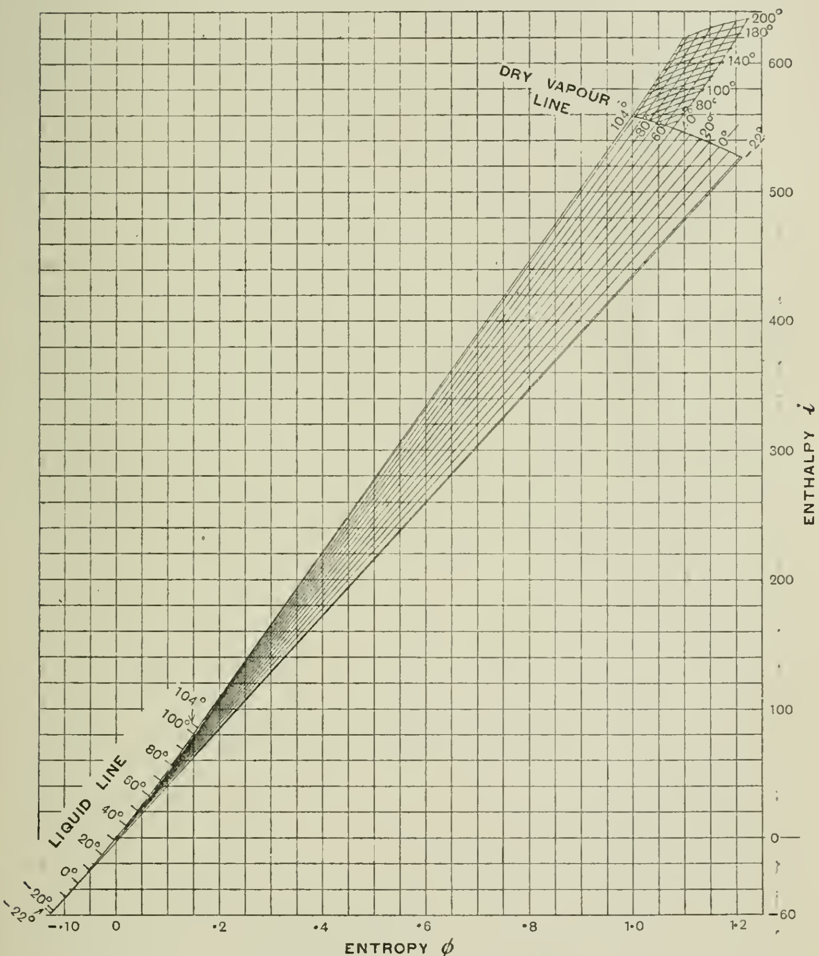



Institution could very well take up and settle, because it was of great importance. Mr. Anderson had also given some Tables, based on Wobsa's 1907 results, but he might not be aware that in the

year 1908 Wobsa published some further figures. The curves of latent heat from both sets of figures would be seen in the diagram.

In April and May 1912 a complete Table of the properties of

Fig. 50.—$\phi$-$i$ *Diagram for* $NH_3$.




ammonia for every degree Fahr. was published in *Ice and Cold Storage*, and he based those on the latest figures by Hýbl; the latent heat curve would be found in the diagram in the middle of the set.

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Dr. Grindley gave a $\phi$-$i$ diagram for $NH_3$ on the basis of skew co-ordinates. Fig. 50 showed a diagram that was in existence, he believed, long before Dr. Grindley's; but it was plotted on squared paper; the Fig. was made from a copy of the original. The alteration in the shape of the curves, when they were put down on a rectangular co-ordinate basis, was worthy of notice. If the frame or border of the diagram shown were considered to be a pin-jointed frame, with the ends of all co-ordinates pin-jointed to the frame, and the bottom left-hand corner and the top right-hand corner were taken and squeezed together, it shortened diagonally in one direction and lengthened diagonally in the other direction, until the skew co-ordinates and Dr. Grindley's curves were obtained.

Mr. Anderson's Paper was full of information suitable for a student's text-book. It seemed rather one-sided, as he believed the makers of the original ammonia machine in England, and the makers of the original $CO_2$ machine in England, were not even mentioned. He quite agreed with the remarks made by the previous speakers in reference to what Mr. Anderson had said with regard to the relation of science to the industry, namely, that no improvements had taken place as a result of scientific investigation. In Mr. Hodsdon's opinion that was not the fact. The author assumed, because no results had been published, that nothing had been done. A buyer could go to any of the leading refrigerating machine makers in the kingdom and get any machine he wanted for any particular purpose. The makers had had to rely on themselves for information; they had not been able to get any from outside sources. They could all deal with the question intelligently; they all had their own testing stations, and their tests were carried out quite independently, because of jealousy between one and the other, and one maker did not know what the other was doing. With reference to the scientific question, the best books on refrigeration and the best monthly papers were undoubtedly published in German, but it was very strange that the three medium-sized refrigerating machines on the largest German liner now being completed were of English make, and were being fitted at the present moment by English engineers. Germany was undoubtedly the home of science, but

science had not helped Germany to get the order for those refrigerating machines.

Fig. 51, Plate 46, was a photograph of the largest $CO_2$ machine in the world, and four machines of the kind shown were running. The gear-wheel shown was 7 feet 3 inches in diameter by 10-inch face, and the motor was of 300 h.p. The compressors were $8\frac{1}{2}$ inches diameter, and the machine would make 70 tons of ice per 24 hours in a semi-tropical climate. The fly-wheel, which was not shown in the photograph, was in halves and weighed 13 tons.

He did not see anything in the Papers about the water-vapour machine, although he believed that several manufacturers were making experiments on such machines. The French Admiralty had several water-vapour machines on the Leblanc system in their ships. It was the latest development of refrigeration, and he had no doubt that they would hear more about it later on. He was sure all the members would be very glad indeed to see the Institution take up research work connected with the subject, but it was necessary to utter a caution before it commenced. The United States Government voted in August of the present year $15,000 to the United States Bureau of Standards for the investigation of the units and standards of refrigeration, and possibly some of the results would be published at the International Congress of Refrigeration to be held in Chicago in September 1913. There were also various Refrigeration Associations which were making independent experiments for the benefit of the refrigeration industry, and some figures had come to hand only that morning of experiments on insulation carried out for the French Association of Refrigeration, particulars of which were given at the meeting held in Toulouse in August last. Units, standard conditions of testing, and similar particulars were also proposed for refrigerating machines. Any work that the Institution did must be done with due regard to, or in conjunction with, what was being carried out by other Associations both on the Continent and in America.

Personally, he would like to see a standard ton of refrigeration. Horse-power did not mean the work that one horse could do, but if the horse-power of an engine was given, an engineer now knew

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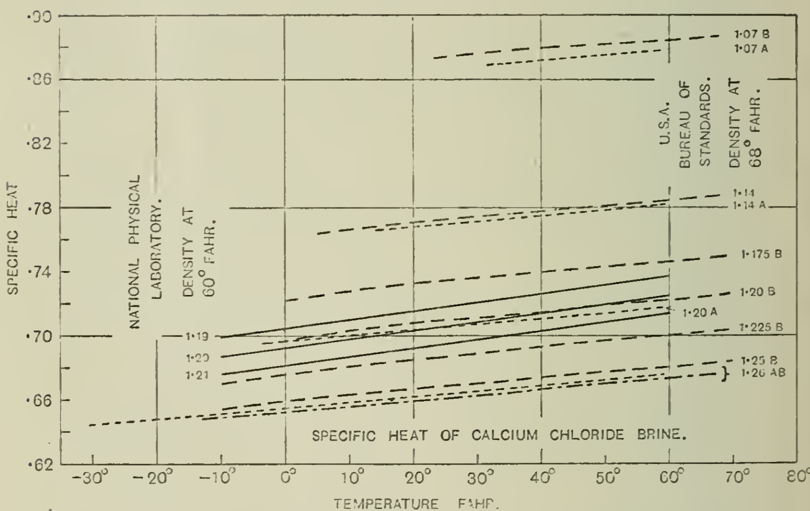
roughly what work it would do. It would therefore be very useful to refrigerating engineers to have a standard ton of refrigeration. It would be necessary to define the amount of duty done and the ranges of temperature between which it would be accomplished.

With regard to point (*d*) at the end of Mr. Anderson's Paper (page 1000), with regard to the charging of the machines, he thought that if a machine were under-charged it stood to reason that the duty would be reduced. If the machine were over-charged the results were perhaps a little doubtful, but the usual result was that

Fig. 52.




a slightly increased duty was obtained at a very greatly increased horse-power, for the reason that the liquid laid up in the condenser and reached the regulator at a temperature nearer the temperature of the condensing water.

Item (*e*) dealt with the question of the value of cooling the liquid refrigerant before expansion in (or by) the evaporator. That dealt with the interchanger question again, and he thought the members would agree with him that there did not seem to be much probability of any gain in that respect. With regard to the question of subjects suitable for investigation, the first in

his opinion was that of the flow of brine through V notches and circular orifices. When a machine was tested, it was desirable to measure the quantity of liquid, either brine or water, passing, and to do that V notches or circular orifices were used. No modern data were available connected with the amount of water or brine that would pass and to what extent the flow was affected by: altering (1) the specific gravity and (2) the temperature of the brine. Reliable investigations were also wanted to determine exactly how the coefficient of discharge for orifices varied with (1) the head and (2) the diameter of the orifice. That was a subject suitable for an investigation, and it could easily be carried out. All engineers possessed certain figures, but it was a point on which agreement was desirable, as the present figures showed great discrepancies.

The next question that required investigation was the specific heat of brine. He did not think there were many refrigerating engineers who knew exactly what the specific heat of brine was. Many figures had been published from time to time, but what was the refrigerating engineer to do when the United States Bureau of Standards and the National Physical Laboratory gave two entirely different sets of figures? He had known refrigerating engineers discuss whether the specific heat of 1·2 specific gravity brine were 0·75 or 0·79 at 0° F., whereas it now appeared to be about 0·69—a difference of 8 and 14 per cent. respectively.

Fig. 52 showed curves from the figures given by the National Physical Laboratory and the United States Bureau of Standards for the specific heat of calcium chloride brine. The series of curves shown appeared to have no connection with one another, but this was altered when they were plotted to a density base as shown in Fig. 53. This diagram also gave a dotted curve recently published by the leading German authority on Refrigeration, as well as the full line curves from the data of the United States Bureau of Standards. It would be seen that the curves approximately passed through the point specific heat 1·0 when the density was 1·0. Owing to the curves being so close together, the rectangle marked A B C D had been enlarged to form Fig. 54, the curves from the data of the

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Fig. 53.—*Comparison of Specific Heats at* 1·20 *Density and* 25·7° *Fahr.*

| | |
|---|---|
| Stetefeld . . . . . . . . . | 0·728 |
| National Physical Laboratory . . . . | 0·716 |
| United States Bureau of Standards . . . | 0·710 |

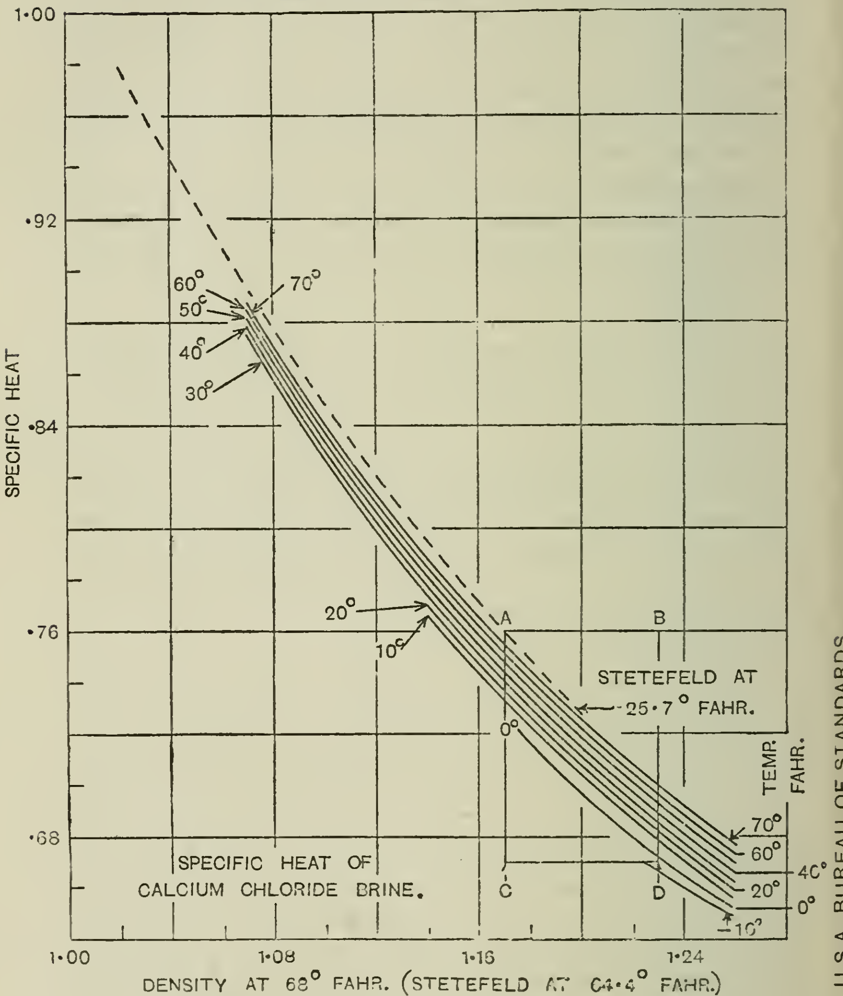
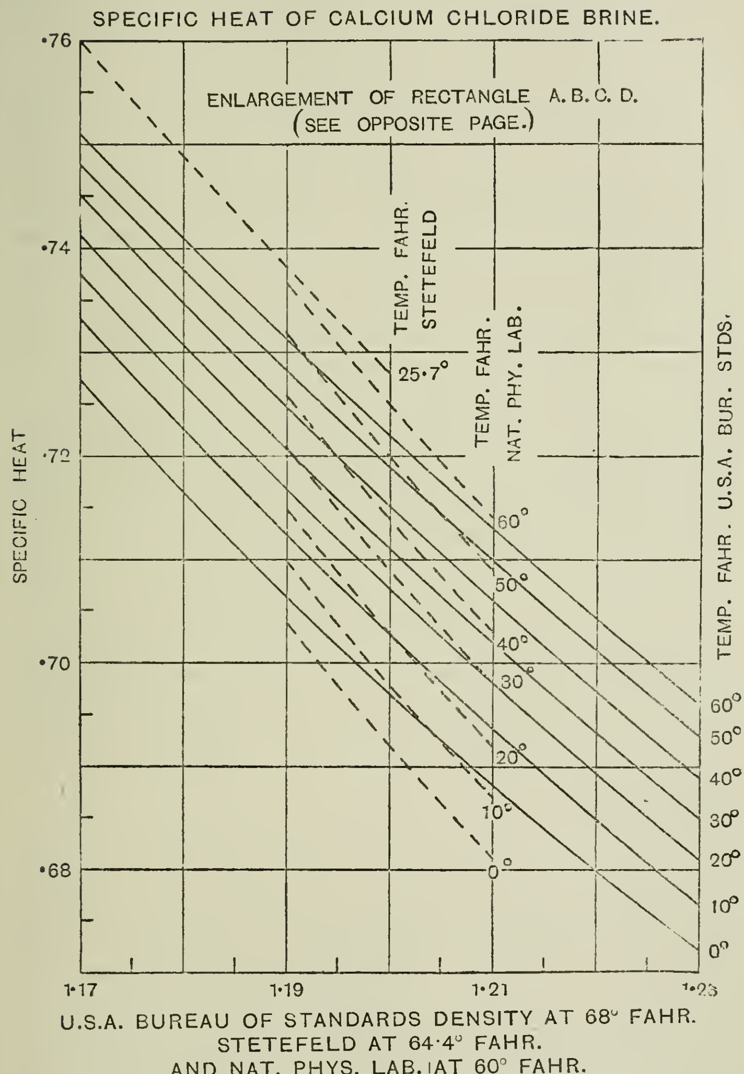


Fig. 54.


SPECIFIC HEAT OF CALCIUM CHLORIDE BRINE.
ENLARGEMENT OF RECTANGLE A. B. C. D.
(SEE OPPOSITE PAGE.)
TEMP. FAHR. STETEFELD
25·7°
TEMP. FAHR. NAT. PHY. LAB.
60°
50°
40°
30°
20°
10°
0°
TEMP. FAHR. U.S.A. BUR. STDS.
60°
50°
40°
30°
20°
10°
0°
SPECIFIC HEAT
·76
·74
·72
·70
·68
1·17
1·19
1·21
1·23
U.S.A. BUREAU OF STANDARDS DENSITY AT 68° FAHR.
STETEFELD AT 64·4° FAHR.
AND NAT. PHYS. LAB. AT 60° FAHR.

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National Physical Laboratory having been added to this in dotted lines. There was certainly no very great discrepancy between the results of any of the authorities, the comparison given at the top of Fig. 53 showing a difference of only 3 per cent. between the lowest and the highest at the particular temperature taken.

Other subjects that required investigation were the Transmission of Heat through Insulation and Heat Transmission through Pipes, the latter of which might well occupy the Research Committee for ten years. It was desirable to ascertain to what extent the transmission was affected by the material, by the pressure, by the temperature, by the kind of gas, by the state of the internal and external surfaces, by the velocity of the gas on the one side and the velocity of the air, water, brine, oil or other liquid on the other side; what were the transmission coefficients for superheated gas to a liquid; what they were for wet gas to a liquid, and what they were for a liquid to a liquid. There were also the questions, of the transmission through brine pipes and direct expansion pipes to rapidly moving air, or air in natural circulation, and to what extent those transmissions were affected by ice or frost on the pipes.

Mr. Henry Brier said that Mr. Anderson gave (page 953) some figures for thermodynamic losses which certainly were not favourable to $CO_2$ machines; in fact they were very unfavourable. It was, however, reassuring to find that in practice these losses did not appear, and that when power and output were taken into account, $CO_2$ machines could compete very favourably with ammonia machines.

The first of the four thermodynamic losses referred to was the loss due to free expansion at the regulating valves. More or less successful attempts had been made to minimize this loss by reducing the temperature of the liquid gas on its exit from the condenser, in order that it might more nearly approach the temperature at which it would be employed in the evaporator. These trials had usually been made either by expanding a portion of the liquid gas itself and returning it direct to the condenser by means of a compressor, or by the employment of

regenerative coolers or interchangers, and many such coolers were at work; but the results up to the present had unfortunately not appealed to the manufacturer or user to any great extent, owing generally to the increased complications of details and multiplication of leaky joints about moving parts, although in many instances much economy had been attained. General attention was so thoroughly concentrated upon this point that one might expect very shortly some practical and, it was to be hoped, entirely successful developments; for, if without extra complications of parts the temperature of the liquid $CO_2$ could be considerably reduced on its passage to the evaporator, it was quite evident that the evaporator would be better able to carry out its allotted work than under the present conditions.

Other losses were referred to, under clauses (*b* and *c*) also on page 953, due to superheat in the compressor and greater pressure given to the gas than the pressure necessary to condense it. Under ordinary or common law one could not expect to get something for nothing, and the concentrated heat due to compression, also the greater pressure given to the gas than was actually required by the temperature of the cooling water for its condensation, was simply payment for a transmission of heat and more rapid liquefaction that would otherwise be obtained. These differences for rapid working were required both in the condenser and also in the evaporator. He was not sure, therefore, that these losses should be counted actually as losses. They were certainly working losses, but were given for something received.

Attention was drawn (page 954) to loss (*d*), and reference was made to controversy between wet and dry refrigeration. Every engineer who had anything to do with the design of refrigerating machinery had had that point before him several years, and had come to the conclusion, in this country at any rate, that the wet gas would transmit more rapidly and with greater working effect than the dry gas. English machines, therefore, were generally worked in the wet state. He believed that some Americans would call it "flooded." It would appear to him (Mr. Brier) quite clear that for a $CO_2$ machine to give its best results, that is to say its

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greatest refrigerating effect, there was a certain point which might be arrived at and be proved by careful trial, and that condition would appear to be when the gas was working in a more or less wet state. It was quite evident that the gas was better able to take up heat from the coils of the evaporator, due entirely to the liquid or wet gas being a better conductor or transmitter of heat, than if it was dry. Another point in favour of the so-called wet process was unquestionably the greater volume of gas which might be circulated by the same swept volume of the compressor piston. Any slightly additional power required to handle this greater quantity of gas would be more than paid for in greater efficiency of the machines, up to a certain point, which was easily obtainable by trial and was well known by all machine makers.

With regard to the question of efficiency on page 956 and also throughout his Paper, Mr. Anderson had classed $CO_2$ as a bad third, and doubtless his figures showed it; but he would like to ask how Mr. Anderson reconciled actual practice with this statement, for, taking his list of powers given on page 1026 and powers in general obtained from other sources of information, one found figures which the $CO_2$ machine could easily pass. He (the speaker) could show $CO_2$ machines which would manufacture in warm countries 36 tons of ice in 24 hours for 84 electrical h.p.; and in speaking of $CO_2$ machines he said he was speaking only of ordinary machines with clearances far in excess of those usually worked on ammonia machines. In the general design of $CO_2$ machines it had been considered that constant reliability of running was of more practical value than absolute efficiency, so that in almost all designs it would be found that whilst the ammonia machine was very often built with suction-valves, which if broken would drop entirely into the cylinder, such a design had never been to his knowledge carried out for $CO_2$ machines. This point was raised simply to show that if $CO_2$ machines were designed in the same risky manner as an ordinary ammonia machine, considerably greater efficiency could be obtained than at present, so that in comparing the powers of one machine with another these points of risk should be taken into consideration.

He (Mr. Brier) was in a position to show that, for ordinary ice-making, the $CO_2$ machine would hold its own in power against the ammonia machine, and he asked the question as to where the falling off took place in the figures credited to the ammonia machines. Surely there was some point which Mr. Anderson had not explained and which was not accounted for in any of the figures he gave. Was it due to better or quicker transmission of the heat under the greater pressure of $CO_2$, or to the greater weight or inertia of the $CO_2$ gas in its passage of the valves? He had often sought an explanation of this point, for he had been constantly told that $CO_2$ (on paper) would not be able to do anything like the same work as ammonia, that $CO_2$ could never bring a cargo of frozen meat across the tropics, that $CO_2$ was dangerous to life, etc. These points, which in practice were never verified, were of course raised by those who at the birth of the $CO_2$ machine held vested interests in ammonia machines.

With regard to the use of copper pipes (page 960), very rarely had he found copper used for $CO_2$ evaporators, and then only in very small machines where lightness or compactness was desired. This point was one of great importance to the marine engineer, and showed how much more suitable $CO_2$ was for marine work than ammonia, owing to the rapid failure of wrought-iron coils if used with sea water.

He wished also to ask Mr. Anderson his reason for stating (page 985) that it was impossible with air circulation to regulate temperatures to a very fine point, and would like to call attention to the fact that attemperated or warm brine was used many years prior to the attemperator referred to on page 996, and was still publicly in use by other firms both on closed and open brine services. There appeared to be a misprint on page 992, in which the figures 35° and 38° F. were given as the temperature of the chilling or attemperated brine. He also desired to know Mr. Anderson's reason for advocating the use of $CO_2$ on passenger ships and ammonia on battleships.

With regard to Mr. Anderson's suggestion for testing vapour-compression refrigerating machines, he would like to suggest that

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most of the points raised in the list given were all points of interest in trials made in almost every machine shop. Needless to say, many of them did not always appear upon the log sheets, but were well known and standardized points, which did not for comparison require repetition in that manner.

With regard to the standard of refrigeration, it would be desirable if the proposed committee would give consideration to the practical side of the question, and if possible to arrange a standard test such as could be carried out in the workshop at a higher temperature than freezing, without the necessity of using brine. If such a standard test or unit could be decided upon, trials could be made with accuracy at much less cost and trouble than if brine were used; at the present moment it was, he thought, the general practice to prove the efficiency of a plant by trial in the workshop, whilst cooling water at 60° F. or thereabouts, with circulating water of the same initial temperature. Pure water circulated gave the most reliable means of directly measuring the number of thermal units eliminated, and was easily obtained from any convenient source at little cost, and had moreover the advantage of leaving the various parts of the plant clean and less liable to damage by rust.

Mr. Arthur G. Enock thought the Papers had provoked an immense amount of discussion which was bound to result in something very useful and practical. It was impossible at the short time at his disposal to go fully into details, and he would therefore confine his remarks to one or two important sides of the refrigerating question. He asked the Members to imagine for a moment the position of a butcher who desired to buy a refrigerating machine to do a certain amount of work for the smallest possible expenditure of power, water, repairs, ammonia or carbonic acid. Such were the businesslike requirements of a shrewd butcher or a cold-storage owner, which had added to the experience and sharpened the wits of refrigerating machine makers. He felt that manufacturers had been rather pilloried by Mr. Anderson, who had accused them of not having made scientific research. He

could show Tables proving that scientific research was being made day by day in many refrigerating machine manufactories, and that the manufacturers of those machines were trying to give the butcher, the dairyman, the ice manufacturer, and the cold-storage owner as many units extracted for as little power and water as possible. They were each trying to do better than the other, and much had been done by careful scientific research in the factories where refrigerating machines were made. He believed that manufacturers in the country could well satisfy the buyer's keenest demands.

Those present had been listening to a number of statements which had been put in relation to each other, when they really had no relation whatever to each other. Some points had been made about dry compression and wet compression, and their suitability for refrigeration or ice-making on a broad scale; while the virtues or vices of the water-jacket had also been placed before them. Those things had their places under certain specified conditions, which existed in a plant at one time and did not exist at another time. For instance, a horizontal double-acting ammonia compressor operating under wet compression would work quite well until the moment arrived when it became necessary to pump out the pipe system, by reason of a joint springing a leak on the ammonia pipe, or from another cause. When the pipe system had to be pumped out, the horizontal double-acting compressor acted for the moment under the wet compression system; but as the ammonia vapour was pumped out from the pipe system, a rarefied gas was gradually obtained. Wet compression did not then exist any longer, but dry compression obtained while the vapour was being extracted from the pipe system, and at such time the double-acting horizontal compressor required a water-jacket. It did not require it while working under the wet compression system, but in working with rarefied gas the piston-rod began to get blue, the packing began to blaze, and a water-jacket was required.

Another condition arising in working cold storage placed the supervising engineer very often in great difficulty with an unjacketed double-acting machine. In working egg-coolers, the

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condition of the air for the best storage results required a fairly high temperature in the air-cooler. Under such conditions, wet gas could not be properly allowed to come back to the compressor, which would heat up unless it were water-jacketed.

On general principles he was of opinion that an ammonia compressor should be so constructed that its working parts, glands, packing, and the like should not require any particular temperature of ammonia gas, but should all work smoothly from a mechanical point of view, whether there were ammonia in the machine or not.

He had made those remarks to illustrate the point that things ought to be discussed in their proper relation. Then with regard to the bursting of a water-jacket, it must be remembered that the water-jacket might burst if it were enclosed, whereas it would not burst if it were not enclosed. Consequently, the vertical machine with the jacket open at the top was a more suitable machine to have a water-jacket than the horizontal machine. Both vertical ammonia and vertical carbonic acid machines were made to work with water-jackets, and operated very successfully.

Mr. Anderson made the following important statement (pages 966–7): "Apart from purely mechanical considerations, one point in the design of a compressor stands out as all important, that is the clearance between the piston and covers at each end of the stroke must be a mimimum—the reason being that the compressor should, if possible, draw on the suction-valve immediately the return stroke begins. The vapour that remains trapped in the clearance space is at the superior pressure of the system, and the piston must travel a very definite distance before this vapour can expand below the suction or inferior pressure, and so allow the fresh charge to be admitted." The speaker found himself in complete agreement with this statement, but was a little surprised, in a Paper on Vapour-Compression Machines, to see that in the types of compressor illustrated and put forward, this important question had been practically ignored. But it should be borne in mind that, while operating without clearance, provision must be made for safe working. It often happened that, instead of only pumping ammonia gas, a compressor was partially pumping liquid.

The compressors illustrated in Figs. 7, 8 and 9 (pages 966–8) would almost certainly break down if liquid flowed into the cylinder along with the gas, and the same would hold good with regard to the machine in Figs. 13 and 14 (pages 973–5). Frequent accidents had happened with compressors of those types. His firm had successfully overcome these defects, and had for the last ten years been making machines which would pass liquid as well as gas without detriment or danger. Such a machine was illustrated in Fig. 55, Plate 46.

In Fig. 55, which represented an enclosed self-oiling machine, the gas came in through the suction stop-valve A down the suction-pipe B and into the pistons C through the holes in upper ends. As the piston descended, gas passed through the suction-valve D and (as the piston ascended) was compressed through the discharge-valve E along the passage F, up the discharge-pipe F (shown partly behind the suction-pipe B), past the discharge stop-valve G into the condenser coils H. The water running over the coils cooled the gas and caused it to condense, and the resulting liquid ammonia passed out through the small pipe J into a receiver (not shown), and was ready to do its cooling work again in the expansion coils (not shown). The crosshead spring K allowed the crankshaft L to press the pistons against the discharge-valve seating V, and thus to expel *all the gas with perfect safety and to pass liquid without danger*.

One also hoped in a Paper on Refrigerating Machines to have seen something about the question of volumetric efficiency and the influence upon it of valve designs. In the compressors illustrated in Figs. 7 to 15 (pages 966–76) the suction and discharge valves were almost all of them side by side in the same end of the machine, and the valve area was greatly restricted. Further, the incoming cool gas was drawn through an annular space in the warm discharge head, and was certain to be wire-drawn and rarefied to a considerable extent. This appeared to the speaker to be bad practice from the volumetric point of view. The machine illustrated in Fig. 55 was provided with a suction-valve in the piston, of a very large diameter, and discharge-valve in the head.

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This design secured a free inflow of gas with a very slight lift of the valve, and a large discharge passage with an equally slight lift of the valve. At the same time, the cold gas came through the cool piston and suction-valve passage instead of through the warm discharge head.

With regard to the question of the unit, a man buying a refrigerating machine wanted to know what he was buying, and the various manufacturers had different ways of describing the units of refrigeration in their machines, and the units of power required per ton of refrigeration. If the sellers of the machines were to try to specify all the conditions precedent to that unit, they would so hopelessly confuse the buyer that he would wish he had never taken up the subject at all. He had with him a Table which, for argument's sake on the horse-power question, showed a variation of 0·7 h.p. per ton of refrigeration to 3·5 per ton of refrigeration—all practical figures taken from actual tests. He commended that point to Mr. Anderson in getting out his unit of refrigeration or power. He also wished to suggest to Mr. Anderson that, in making any scientific research on the subject, he should get every type of machine installed under similar conditions, and then ascertain what was the value of clearance or no clearance, what was the real value of water-jacket or no water-jacket, and what was the value of wet compression or dry compression, also what type of compressor possessed the highest volumetric efficiency. Mr. Anderson would find that any manufacturer in this country, who was enterprising enough to make known what he was doing, would let him test his machines in his works with electrical and other appliances to determine the value of those points.

Mr. John Thom desired to make a few remarks, not as a manufacturer but as one who had had to deal with the running of refrigerating machines of various makes during the last twenty years. Going back to that time, there were very few machines in use except those of the cold-air class. When machines of the vapour-compression class were introduced, the difference in the power required to drive them was about in the proportion of

1 ton of coals to 5 tons with the air-compressor, which was a radical improvement. When refrigerating machines were first put on board ship, they were installed as a sort of auxiliary after all the other machines had been put on board, and were run with the same boilers. He (Mr. Thom) remembered well the extraordinary number of spare parts, etc., thought necessary for a refrigerating machine at that time, also that there was a spare machine provided for doing the same amount of work. Vapour machines existed twenty years ago, but they were few in number, and shipowners were rather afraid of fitting them, in case of damage to cargo or passengers, but it was very pleasant to hear even at that time the difference in the amount of steam consumed in the one machine compared with the other, namely, about as 5 to 1. He had had a few of the different kinds of vapour machines under his care for some years, some of which had been very satisfactory, and had worked with a very small coal consumption, while others had been the reverse. The machines which had given him the most trouble were not the $NH_3$ nor the $CO_2$, but the $SO_2$. In a machine of the $SO_2$ kind which had been running under his charge for four or five years, some of the tubes in the evaporator had been entirely closed. [Mr. Thom then handed in a sample of the tube.]

Most of the speakers in the discussion were manufacturers of refrigerating machines, and they all seemed anxious to show that their firms spent time and money on the scientific side of refrigeration, and that every one of their machines would give the very best results. He (Mr. Thom) remembered having a machine under his charge in the manufacture of which very little science had been bestowed, not as regards its principle of working, but in the manufacturing and proportioning of the article. The machine had to be stopped every few days, pulled to pieces, and adjusted in some way or other to prevent it breaking itself to pieces. What was the good of science if the constructor did not carry out his part properly? It was all very well talking about science in a drawing office and before such a Meeting as the present, but science was no good unless it was practically applied to the strength of materials used in the construction of one's machines, and which

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would do the work without becoming deformed. The $SO_2$ machine was a very economical refrigerating machine in theory and if the evaporator were worked above atmospheric pressure, but not if it were worked below atmospheric pressure. The speaker's experience with these machines had been most unsatisfactory, through getting moisture into the system at glands, etc., which in time caused the evaporator to fur up; this did not happen with any of the other refrigerants. Moreover, $SO_2$ as a chemical did not seem to be made so pure as $NH_3$ or $CO_2$.

On page 963 an $NH_3$ condenser was shown with about 70 connections on each section, which in these days of electric welding seemed out of date. He thought that a condenser, or any arrangement on a refrigerating machine having so many joints as this, was the worst kind one could have. Many of the joints were screwed and soldered at the makers' works, but there was the possibility of them leaking under peculiar circumstances, such as rough handling in getting the tubes from the manufacturer to the place at which they were going to be fitted up; consequently they would give trouble in course of time. Personally, he thought this was an out-of-date class of condenser. A condenser with four or five joints per section would be much more reasonable.

The compressor shown (page 968) had a delivery valve with a very large amount of clearance in comparison with the suction ones. On page 970 there was shown a 12-inch × 21-inch triplex $NH_3$ compressor driven direct from the tail-rods of the steam-cylinders; this tail-rod connection with vapour compressors added very considerably to the clearance of the compressors, as due allowance had to be made in the compressor-cylinder for the expanding or contracting of the steam cylinder-rods. He (Mr. Thom) always advocated compressors being connected direct to the crankshaft.

Mr. Cracknell, who spoke on the absorption machine (page 1067), gave them the figures of $15\frac{1}{2}$ tons of ice per ton of coal in special cases and 14 tons of ice per ton of coal as common with the absorption machine. Then he went on to say the compression machine made only 13 tons of ice per ton of coal. Mr. Thom was afraid that Mr. Cracknell selected a rather uneconomical plant for

comparison. He himself knew of large compression plants making as much as 40 tons of ice per ton of coal with gas-engine drive, and similar size plants steam-driven, making regularly 23 to 25 tons of ice per ton of coal.

In making comparisons between $NH_3$ and $CO_2$ machines, the latter reached the critical temperature in the tropics, the former not so; engineers experienced this when they had condensing water exceeding 85° F. The author spoke of $NH_3$ machines in battleships, and advised blowing off the charge before going into action. If one did that, how was the magazine to be kept cool? If this was necessary, he would advise fitting $CO_2$ machines in place of $NH_3$ machines to all battleships.

A unit of refrigeration, if it could possibly be arranged, would be an excellent thing for the man who bought a refrigerating machine and also the man who had to look after it. It would be an advantage, not simply from the point of view of marketing only, but as a method of comparison. If this work could be undertaken by the Institution in conjunction with manufacturers, scientists and those who actually ran refrigerating machines, much good would result.

Mr. Anderson, in reply, said it was evident that Professor Jenkin had given considerable thought and attention to the subject of refrigeration, and one could but look forward with interest to the publication of the results of his investigations. No one could possibly question the great advantage of the Centigrade thermometer scale, but as he (the author) had placed one of the two outstanding wants in refrigeration as a "standard ton of refrigeration," it seemed quite out of place to suggest anything so revolutionary as a new thermal unit displacing the British thermal unit. As he had mentioned on page 998, the United States had adopted a standard ton of refrigeration, and it had been seriously suggested that this country should adopt the same unit, namely, 288,000 B.Th.U. per day per ton of refrigeration. That meant—for the purpose of that unit at any rate—adopting the 2,000-lb. ton. Any committee appointed to consider the question of units and standards in

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refrigeration would probably have those and many other suggestions before them.

Mr. Voorhees, like Mr. Cracknell, questioned his statement with respect to the ammonia absorption machine. In 1906 he had the advantage of a tour in the United States, and was very much impressed by the number and the size and efficiency of the absorption machines he saw at work. If it was simply a question of turning out so many pounds of ice per pound of coal, he would hesitate to place the compression before the absorption machine. Taking, however, the many points which had to be considered when dealing with refrigeration—such as meeting overloads and underloads, and such questions as power supply, etc.—he had no hesitation in saying that the compression machine was destined to keep its place, as being by far the most commercially important of all types, but he was as equally sure that absorption machines had not received the attention in this country that their efficiency undoubtedly merited.

Mr. Voorhees further stated (page 1058) that he (the author) had overlooked the question of superheating in the compressor, and its effect on the volumetric efficiency. As a matter of fact he had not by any means overlooked that important point. On page 1021—the page referred to by Mr. Voorhees—he said in the second line, "and allowing for losses at the regulating valve, influx of heat into the pipe connections and cold parts, and by superheating in the compressor, 450 B.Th.U. per pound of ammonia circulated may be taken as available refrigerating effect." In so long a Paper it must have been that Mr. Voorhees had overlooked those words, and his attention arrested by the 9·1 cubic feet lower down. To make allowance for superheating, he (Mr. Voorhees) would increase that value to about 10·7 cubic feet. If that were done, the value 450 B.Th.U. would need to be increased, and the value of the fraction which was given in the centre of page 1021 would in consequence remain exactly the same. Personally, he preferred to make allowances for all estimated losses at one operation, and the figures as given were perfectly correct and the size of the compressor need not be increased by 17½ per cent. as Mr. Voorhees suggested,

He thanked Mr. Voorhees for his valuable contribution to the discussion.

With regard to the remarks of Mr. Willcox, he knew the leading makers of refrigerating machinery kept thoroughly abreast of their work in all departments—scientific and practical—in fact, as he had written, "quite in keeping with the best traditions of the British engineer." Nevertheless he must emphasize what he had written at the top of page 950, that no branch of mechanical science had received less aid in this country from published research or from accounts of practical progress than that of mechanical refrigeration. He did not realize how poor this country was in the literature of the subject until he set to work on the Paper. Other countries had certainly more literature, but the bewildering conflict of results and opinions led him to think that in this country they were better off with a comparatively clean slate. An immense amount of work remained to be done, and he was glad of the support of Professor Jenkin in his opinion that one of the greatest needs was reliable data as to the refrigerants themselves. When Mr. Willcox was trying to persuade the Institution that no further research was necessary, he (the author) remembered the fact mentioned by Mr. Hodsdon that the U.S.A. Congress had just voted $15,000 "for investigation incident to the establishment of units and standards of refrigeration and the determination of the physical constants of materials used in the refrigeration industries, such as ammonia, aqueous ammonia solutions, carbonic acid, brines, etc." The request for that vote came from the American Association of Refrigeration, and their petition included the words, "the work of this Association and the entire refrigerating industry has been hampered from its inception from lack of definite basic standards upon which the art of refrigeration is based." *Ice and Refrigeration* for September 1912, in commenting on that grant, said, "It marks a distinctive step in the advancement of the industry toward higher achievement made possible through the use of officially authenticated basic data upon which refrigerating engineers of the world may rely." He reminded the members that his Paper as printed was finished in February 1912, and the vote to which reference was now made was taken in September 1912.

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As to Mr. Willcox's question as to what was meant by "a standard refrigerating machine of comparison," he would say it was a standard machine by which the efficiency (coefficient of performance) of other machines could be compared at any time and under any conditions. The standard engine of comparison for reciprocating steam-engines was one working upon the Rankine cycle, A B C D, Fig. 34 (page 1011). A possible standard refrigerating machine was the reverse of that, namely, A D C B. Since the Paper was read, and having Mr. Willcox's question in mind, Mr. G. W. Daniels, an engineering student of the University of Liverpool, had plotted to scale a diagram similar to Fig. 34 for an ammonia machine working between 14° F. and 68° F., and had found that a machine working on the ordinary cycle, approximately F D C B, Fig. 34, showed a falling off from the reversed Rankine cycle, A D C B, of 6·7 per cent. That indicated how a comparison might be made, and incidentally showed the value of undercooling. It might be further remarked that a standard machine of comparison would make the system of heating the condensing water to a specified temperature during a test quite unnecessary.

Captain Sankey would notice that he (the author) had mentioned the standard machine of comparison on page 1000 and nowhere else. He agreed entirely with the figures given by Captain Sankey (page 1070), who had made one of the most valuable contributions to the discussion. There were many different cycles on which a machine might work, and one or two of them had been suggested in the course of the discussion, while one in particular had lately been prominently put before the public by Mr. Voorhees and Mr. Stokes (*see* Communication by Mr. Stokes, page 1130), while there was also Dr. Grindley's cycle. It was, perhaps, just as well that he had not gone any further into the suggested standard machine, because all these different cycles would have to be very carefully thought out before a definite cycle could be fixed upon.

With reference to Mr. Harrap's remarks (page 1071), he would like to refer particularly to the work done by Mr. T. B. Lightfoot, whose very valuable Paper was written in 1886. He (the author) took the work up from the point left by Mr. Lightfoot, and those

who read that Paper carefully would note that the advancement had not been by any means so marked as some of the speakers would like to make out. He had not in any way attempted any of the work that Mr. Lightfoot had covered so well, and anyone studying the subject of refrigeration should not fail to read the earlier Paper. Another point mentioned by Mr. Harrap was that it was possible to theorize as much as one liked, but when it came to practice very large allowances had to be made. He had in Appendix V (page 1021) given the barest outline of the design of an ammonia machine, and in the columns and Tables that were given on page 1027 the great amount that had to be allowed for leakage of heat into the machine would be noticed. He thought that those who took up the testing of refrigerating machines often overlooked that very important point, and sometimes lost heart because their heat balances would not work out.

He had to thank Mr. Tyler very much for his contribution to the discussion (page 1075), one of the main points he mentioned being that of rating. The Americans had adopted the figure of 12,000 (page 998), the same as he (Mr. Tyler) had. When a machine was rated at so many tons of refrigeration it might perform, say $x$ tons of refrigeration under certain conditions, and $y$ tons under other conditions. What was required was a standard whereby it could be said that the machine under standard conditions of temperature would be rated at a definite amount; and then if it was used for cooling milk it would eliminate a larger number of thermal units per day, because its coefficient of performance would be very much higher. If, on the other hand, it was making ice it would be correspondingly less. It did not really matter whether the temperature was 32° or 0° so long as the standard was fixed, but Mr. Tyler's suggestion of 32° F. was certainly worthy of consideration, but in any case an upper limit would also be required, say 70°, 80° or 90° F. He liked Mr. Tyler's term—"a frigid ton." "A ton of refrigeration" was, as Mr. Tyler said, rather ponderous, and it was just possible that if a Committee were formed, some short term might be decided upon. Mr. Tyler's idea of making the uses practically the same for the parts shown on the

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diagram given on page 952 was quite a good one from a practical point of view, but in that case it had to be borne in mind that the temperatures and pressures would be reversed. As it was at present, the temperatures and pressures were kept in each vessel relatively the same, while the cycle traced out was the reversed Rankine cycle, and from a theoretical point of view this had much to recommend it. He agreed entirely with Mr. Tyler regarding wet and dry compression. Personally, he believed that the $CO_2$ machine was better worked rather on the wet side, the $NH_3$ machine a little on the dry side, and the $SO_2$ machine decidedly dry.

With regard to Mr. Hodsdon's remarks (page 1080), personally he very much regretted that Messrs. Linde and Messrs. Hall found it impossible to let him have some diagrams of their machines, and the Paper for that reason was not perhaps as representative as it might have been, but the other makers had kindly supplied all that was necessary.

Mr. Brier (page 1094) said that the author had referred unfavourably to $CO_2$ machines, placing $CO_2$ a very bad third. This was, of course, theoretically. On page 956 he had used the words "here again really reliable comparative tests are wanting," and also the words "*if* reliable tests proved $CO_2$ to be as inefficient as theory points out," while on page 1028 he had written "that much study and careful research is required to put the knowledge of this class of machine on a satisfactory basis." This was the crux of the situation; theory pointed one way, practice apparently another, and surely these were ideal conditions for research, which he would again urge was greatly needed.

The President reminded him the usual time for adjournment had long been exceeded, and in consequence he would reply in writing to the other points that had been raised (*see* page 1135).

*Communications.*

Mr. Alexander E. Beck wrote that the "double-pipe" condenser, described and illustrated on pages 964–5, had a multiplicity of joints which increased the chances of loss by leakage, and he thought that if it were exposed to a moist or damp atmosphere and neglect, surely corrosion would make it difficult to effect repairs. If Mr. Anderson could give the result of, say, four or five years' wear and tear, it would be of interest to many.

Fig. 19 (page 981) illustrated a typical milk-cooling plant as used by dairymen, and with regard to this class of refrigeration, with which the writer had had considerable experience, the point to be considered was that dairymen required quick action, whether they pasteurized or not. It seemed a very simple problem to cool at the rate of 200 or 300 gallons of milk per hour from, say, 70° to 40° F.; this was not so, and by manufacturers of refrigerating machinery the duty of the refrigerating plant was often under-estimated. In dealing with either large or small quantities of milk, it was essential to have a fair quantity of the cooling medium in circulation; otherwise a larger refrigerating plant was necessary. Some twenty years ago this fact was very forcibly brought to the notice of his firm when they devised the "storage system," which consisted of an elevated brine storage-tank from which the cold brine was led to the milk cooler and flowed by gravity back to the evaporator or a lower brine tank of equal capacity to the elevated tank. By this means the cooling medium might be cooled by the refrigerating plant preparatory to cooling the milk, and thereby a smaller plant employed effectively for the purpose.

On page 1021, 4·53 cubic feet of compressor displacement per minute per ton of refrigeration per day was given. The writer presumed this should be multiplied by 2 for ice making (roughly), which would be, say, 9 cubic feet per minute per ton of ice per day. He would refer to the result of an analysis of displacements

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provided for by six firms, who submitted estimates for the supply of plant to make 15 tons of ice per 24 hours. They were as follows:—

| | | |
|---|---|---|
| No. 1 | 10·2 | cubic feet displacement. |
| No. 2 | 13·2 | ,, ,, ,, |
| No. 3 | 12·15 | ,, ,, ,, |
| No. 4 | 13·0 | ,, ,, ,, |
| No. 5 | 15·4 | ,, ,, ,, |
| No. 6 | 12·73 | ,, ,, ,, |

The condensing water was specified as 80° F., and there was an abundance of it. No. 5 machine exceeded the specified duty by 5 per cent. on test runs. The area of condenser surface per ton per 24 hours averaged from 70 to 80 square feet, and area of evaporator surface about 80 square feet per ton per 24 hours. Comparison with these figures seemed to indicate that too little was allowed for compressor displacement by Mr. Anderson, even having regard to the temperature taken as a basis.

Mr. E. J. Buckton (Tilbury Docks) thought the present rapid growth of cold storage would be accompanied by many changes and developments of refrigerating plants by competing manufacturers both at home and abroad. As the number of classes of goods to be preserved increased, specially designed chambers would be brought into use and cooling systems would be devised to suit the requirements of each kind of merchandise and local conditions; and the types of cold stores were likely to multiply in number rather than decrease. On the other hand, only a few types of compressors were really necessary to meet all possible demands, and the tendency should be towards standardization.

Horizontal compressors had been extensively used for years, and, as Mr. Anderson had pointed out, they were to a certain extent standardized. There were many successful vertical compressors in use, but the manufacture of the vertical type had been lately discontinued by at least one important firm in favour of the horizontal compressor. In the past, most large compressors had been driven by an independent plant, usually steam, but the

majority of cold stores were situated at important ports and in large towns, where in these days electric current was obtainable at rates sufficiently low to have brought about the supplanting of steam-engines by electric motors, and many compressor stations had recently been converted, usually by disconnecting the direct steam-drive and substituting a belt-drive from a separate motor to the fly-wheel. Chain-gears with spring wheels had also been used, but not always with complete success.

At the present time, where electric power was available, compressors other than very small ones were usually specified to be belt-driven. Slow-speed direct-coupled motors were, in some cases, used for small powers, and a number of horizontal compressors had been built with a helical pinion gearing with the fly-wheel, but it was doubtful if this drive would be adopted to any extent for very large horizontal machines, unless the present designs were modified to give a more uniform crank-effort diagram. It would be interesting to know how, during the next few years, manufacturers would meet these altered conditions.

With a geared motor drive, the higher the compressor revolutions the better. This favoured a return to the vertical short-stroke type, but it was probable that the stroke would be shorter relative to the diameter than formerly and the revolutions greater. Also, the higher powers would be obtained by increasing the number rather than the dimensions of the cylinders. Mr. Anderson had spoken of the advantage of reducing the clearance to a minimum, and this was most important when the compressor cylinder was short and single acting. There were several mechanical contrivances in use for giving no clearance, which were worthy of careful investigation. Up to the present, the horizontal double-acting compressor had more than held its own, and in England the larger plants were almost exclusively of this type; but the existence of central electric power stations in the large towns might bring about important changes in compressor design, and it remained to be seen if a multiple-cylinder, high-revolution type of geared machine would oust the present partially standardized horizontal, long stroke, belt-driven machine from its present position.

Mr. W. S. Douglas wrote that he thought his firm (Messrs. William Douglas and Sons, Putney) was the only one in this country making $SO_2$ machinery, and that the following remarks might not be inappropriate.

Dr. Grindley's speculation was interesting, particularly from the mathematical thoroughness with which he had followed it through, but, as he said himself, the principal benefit would be to $CO_2$ machines. $SO_2$ and $NH_3$ machines would reap a very much smaller theoretical, and probably a negligible practical, advantage from the cycle. In any case, the writer's own experience was that the most efficient of water-jackets, even with an ample supply of cold condensing water, still left one very far from isothermal compression. The heat of compression was generated so quickly that the cylinder walls could not transmit it in the time. He was afraid therefore that the cycle, except in a very imperfect form, would be impracticable.

With regard to Mr. Anderson's Paper, he (the writer) was personally quite in favour of public research work, and would welcome the standardization of the refrigerating ton, which should be a round figure, for the purpose of simplifying calculations; he hoped that the temperatures or pressures of evaporation and condensation would be clearly fixed. He would also like to add to Mr. Anderson's subjects for research work the Transmission of Heat across Coil Surfaces. No exact laws had so far been formulated, except in the roughest and most empirical fashion, to govern this transmission, as far as concerned the refrigerating engineer, and the laws governing heat transmission in steam-pipes, etc., were found not to apply. He would also suggest the standardization of the physical data for each refrigerant both in Fahrenheit and in Centigrade units, and the publication of standard Tables.

With reference to Mr. Anderson's remarks concerning $SO_2$ machines, the writer would like to add one more to the advantages which these machines possessed. This was, that liquid $SO_2$ was a lubricant and that consequently, if the cylinder of an $SO_2$ machine were water-jacketed, it was quite unnecessary to lubricate its interior. It followed that lubricating devices, oil separators,

rectifiers, etc., were not required with this type of machine, and its construction was thus considerably simplified. He noticed that Mr. Thom (page 1101), speaking as a user, said that the $SO_2$ machine gave him most trouble. He referred, no doubt, to German, or at any rate to foreign machines. If he were to try a British-made $SO_2$ plant, the writer was certain that Mr. Thom's opinion would be exactly reversed. The writer's firm now manufactured $SO_2$ machines, and had at one time, and for a number of years, sold $CO_2$ machines of another make. It had been their experience that $SO_2$ as a gas was much more easy to deal with than $CO_2$, and that far less trouble was incurred both in running and in manufacturing the machines.

Mr. Thom also stated that $SO_2$ machines did not work well below atmospheric pressure. Probably this experience was gathered from the machines above referred to, but the writer's experience had been quite the contrary. Air, of course, got into $SO_2$ machines, as well as into others, but in no greater quantities and with no worse results. In this connection it was well to remember that, with the gas evaporating at 0° F., the vacuum required was only 4 lb. below atmosphere—a difference in pressure which could hardly be called dangerous. His own firm would be most pleased to take part in a conference to discuss a British Standard ton of refrigeration, should such a conference be called.

Mr. Joseph Hill (Sheerness Water Works) wrote that he considered Mr. Anderson's Paper to be the best on this subject since Professor (now Sir James) Ewing delivered his lectures in 1896–7. Dr. Grindley's Paper was of great interest in that it showed how more work could be got from the refrigerant. The chief difficulty lay in having to deal with a dry gas, which would of course be fatal to machines of the Linde type, owing to trouble with the glands. No water-jacketing could be effective in practice, the heat not being passed quickly enough through the cylinder walls. If the new cycle were tried, it would be advisable to use a machine built on the De La Vergne system with oil injection and a long stroke slow speed.

Mr. Robert Knox wrote that he was personally interested in the subject of refrigeration, having been engineer for three years to an ice company in the tropics, and as the working conditions in hot countries were usually very different from those obtaining in more temperate climes, his remarks might perhaps be of interest. In Colombo, Ceylon, he had had charge of refrigerating plant comprising three separate units, making 2, 6, and 18 tons of ice per day respectively, giving a total output of 26 tons of ice per day of 24 hours. All the units were steam driven, and the refrigerating agent employed was sulphurous anhydride ($SO_2$). There were three other ice factories in Colombo, all having "ammonia" machines, and as regards the merits of the one refrigerating agent over the other, he believed that $SO_2$ was more suitable for the tropics on account of the lower pressure obtained with it in the compressor cylinder of the plant, as compared with the higher pressure obtained when ammonia was used.

One drawback with $SO_2$ was the danger of air leakage when the temperature of the brine fell so far as to bring the $SO_2$ in the suction side of the machine under atmospheric pressure. With ordinary care and supervision, however, there was little danger of an air leak, but when a leak did occur the consequences were unfortunately not detected for some time, and only then by noticing the increased pressure in the compressor-cylinder and the failure of the plant to lower the temperature of the brine by the usual amount in a given time. A small leak on the pressure side of the plant was as a rule readily found by testing with litmus paper, but it was not so easy to locate a leak on the suction side, because, while the plant was working, air would be drawn in, and before a leak could be found, it was usually necessary to stop the plant and investigate. In spite of the chance of this occasional trouble, however, he believed that the $SO_2$ machine was the best for hot countries.

As a whole, the plant under the writer's charge was satisfactory in working, but the two smaller units were more economical than the larger one. In this connection he thought that designers of refrigerating plant did not fully appreciate the conditions obtaining in the tropics, as although a small unit would work with an

economy only slightly less than with the same machine working at home, it would appear from his own experience that the larger the machine the greater the drop would be in output as compared with its performance at home. For example: The 2-ton machine was always up to time with its ice, turning out 2 tons every 24 hours as regularly as the clock. The 6-ton unit was not quite so efficient, and required as a rule longer time, while the 18-ton machine was still less efficient, and usually required from $1\frac{1}{2}$ to 3 hours over the 24 to make its full rated output of ice. This performance of the large machine was unsatisfactory, because it was necessary for the business to have the ice ready for lifting at 6 a.m. every morning, and this requirement was only met by reducing the output of the plant by 4 or 5 cwt. of ice per day.

He attributed the relatively unsatisfactory working of ice plant in the tropics as compared with identical plant at home to three causes, namely :—

First, the attendants were usually natives of the country, and although their capabilities were fairly good for running a steam-plant, as would be expected they were not so intelligent as the men at home when put to watch a refrigerating plant. In the writer's opinion, such plant was the most difficult plant to work so as to get continuously good results, and as so much of its efficiency depended on the correct setting every hour or so of the valve regulating the flow of the refrigerating agent from the condenser back to the evaporator, it required an attendant of more than the average native intelligence. When it was understood that ice-making plant in the tropics usually ran day and night for weeks without a stop, and that it was not possible for the European engineer in charge to be on duty all the time, it would be seen that many things might happen which were detrimental to the efficiency of the plant.

The second cause was the much higher initial temperature of the water in the tropics from which ice was to be made, as compared with the temperature of the water at home, and this, coupled with the higher temperature of the atmosphere of warm climes, put a greater load on a machine working under such

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conditions as compared with an identical plant at home; consequently there was a drop in efficiency and output due to this.

The third cause, and the one which he found to tell most against the performance of ice plant in the tropics, was the extremely high temperature of the condensing water available. It would perhaps astonish those engaged in designing ice plant to see from the following Tables of observations (page 1118) that at times the temperature of the water at the condenser inlet was as high as 36° C. (96·8° F.). Had there been running water near to the factory, the temperature would never have been so high, but in common with the three other factories in Colombo they had to use the water of an inland lake which was very shallow, had much decaying vegetable matter in its water, and was heated all day by a tropical sun.

Matters like the three just mentioned were sometimes overlooked by designers of ice plant at home, and also unfortunately by those requiring plant for the tropics, and it could not be too carefully kept in mind that a plant designed for home service, or on a wrong knowledge of the conditions obtaining in hot countries, would almost certainly be inefficient, due to any or all of the causes he had stated.

The 18-ton machine with which the writer had to do was designed by the makers to suit a temperature of condensing water of 30° C. (86° F.), and the engine power was fixed at 70 h.p., but as the maximum temperature of the condensing water rose at times to 36° C. (96·8° F.), it was found that 100 h.p. was required for the working of the plant. The makers' statement, made after this plant had been put to work, was as follows: "In all ice machines, of whatever type they may be, the necessary power required increases by 5 per cent. for each degree (Centigrade) of the condensing water above 30° C." The condensing water was often 6° C. above the figure mentioned by the makers, and the extra power which was required over the rated power of 70 h.p. approximated to the amount estimated in accordance with the makers' statement, namely, about 30 per cent. due to the condensing

water being at 36° C. instead of 30° C. As regards the quantity of condensing water required, the makers said: "The quantity of condensing water must increase as the temperature increases, and for the 18-ton plant, if the temperature is 34° C. the minimum quantity of condensing water should be 14,000 litres per hour, with a difference of 5° C. from the inlet to the outlet." As a matter of fact, the maximum amount of condensing water which could be used for the plant in question was fully 40 gallons per minute, and this at times at a temperature of 36° C., as against 51 gallons per minute, that is, 14,000 litres per hour stated by the makers as being required for water of 34° C.; as would be seen from the Tables of observations the difference in temperature was at times as much as 10° C. from the inlet to the outlet.

From the foregoing it would be evident that the 18-ton machine was inefficient, and this instance emphasized the importance of getting absolutely reliable data before designing plant for service in hot countries. The three ice machines of which the writer had charge were made by a French firm, and they were all of the horizontal type, having their compressor set tandem behind the steam-cylinder. The main particulars of the largest unit, namely, the 18-ton machine, were as follows:—

| | |
|---|---|
| Steam cylinder . . . . | $16\frac{3}{4}$ in. diameter × 2 ft. $7\frac{1}{2}$ in. stroke. |
| $SO_2$ compressor . . . . | About $15\frac{1}{2}$ in. diameter × 2 ft. $7\frac{1}{2}$ in. stroke. |
| Revolutions per minute . . | 75. |
| Boiler steam-pressure . . | $6\frac{1}{2}$ atmospheres. |
| Temperature of feed-water . . | 140° F. |
| Vacuum in steam-condenser . | About 22 in. |
| Maximum estimated rated h.p. . | 70. |
| Actual h.p. at maximum load . | 100. |

The following tabulated observations gave several readings which might be of interest as showing the working conditions of the 18-ton machine:—

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*Colombo, 29th Oct.* 1901.

| Time. | Condensing water for $SO_2$ Condenser. | | | Temperature of brine in ice tank. | Readings on compressor gauges. | | Position of $SO_2$ regulating valve. |
|---|---|---|---|---|---|---|---|
| | Inlet Temp. | Outlet Temp. | Quantity per min. | | Suction side. | Pressure side. | |
| | °C. | °C. | Gal. | °C. | Atmospheres. | | |
| a.m. | | | | | | | |
| 11. 0 | 32 | 43 | 40·5 | −1 | 0 | 6·5 | ½ turn open |
| 11.30 | 33 | 44 | ,, | −1·5 | 0 | 6·5 | ,, ,, ,, |
| p.m. | | | | | | | |
| 12. 0 | 34 | 44 | ,, | −1·75 | −0·1 | 6·6 | more than ,, ,, ,, |
| 12.30 | 34·5 | 44·5 | ,, | −2 | −0·1 | 6·75 | ¾ ,, ,, |
| 1. 0 | 34·5 | 44·5 | ,, | −2·25 | −0·15 | 6·8 | ,, ,, ,, |
| 1.30 | 35 | 45 | ,, | −2·25 | −0·15 | 7 | more than ,, ,, ,, |
| 2. 0 | 35 | 46 | ,, | −2·25 | −0·2 | 7 | ,, ,, ,, ,, ,, |
| 2.30 | 36 | 46 | ,, | −2·5 | −0·2 | 7 | ,, ,, ,, ,, ,, |
| 3. 0 | 35 | 46 | ,, | −2·5 | −0·2 | 7 | ,, ,, ,, |
| 3.30 | 34 | 44·5 | ,, | −3 | −0·2 | 6·8 | ,, ,, ,, |
| 4. 0 | 33·5 | 44·5 | ,, | −3·25 | −0·2 | 6·8 | ,, ,, ,, |

*Colombo, 4th Nov.* 1901.

| Time. | Condensing water for $SO_2$ Condenser. | | | Temperature of brine in ice tank. | Readings on compressor gauges. | | Speed of engine. Revs. per min. | Position of $SO_2$ regulating valve. |
|---|---|---|---|---|---|---|---|---|
| | Inlet Temp. | Outlet Temp. | Quantity per min. | | Suction side. | Pressure side. | | |
| | °C. | °C. | Gal. | °C. | Atmospheres. | | | |
| a.m. | | | | | | | | |
| 10. 0 | 30·5 | 38 | 40·5 | −0·5 | −0·1 | 5·75 | 71 | ½ turn open |
| 10.30 | 30·5 | 39 | ,, | −1 | −0·1 | 6 | 71 | ,, ,, ,, |
| 11. 0 | 32 | 39·5 | ,, | −1·75 | −0·15 | 6 | 71 | ,, ,, ,, |
| 11.30 | 32·5 | 40 | ,, | −2 | −0·15 | 6 | 72 | ⅝ ,, ,, |
| p.m. | | | | | | | | |
| 12. 0 | 32·75 | 40·5 | ,, | −2 | −0·2 | 6·125 | 73 | ,, ,, ,, |
| 12.30 | 33·5 | 41 | ,, | −2 | −0·2 | 6·25 | 72 | ,, ,, ,, |
| 1. 0 | 34 | 41·5 | ,, | −2·125 | −0·2 | 6·25 | 71 | ,, ,, ,, |
| 1.30 | 34 | 41·5 | ,, | −2·125 | −0·2 | 6·375 | 72 | ,, ,, ,, |
| 2. 0 | 34·75 | 42 | ,, | −2·25 | −0·2 | 6·375 | 73 | ¾ ,, ,, |
| 2.30 | 34·75 | 42·5 | ,, | −2·5 | −0·2 | 6·75 | 72 | ,, ,, ,, |
| 3. 0 | 34·25 | 41·75 | ,, | −2·75 | −0·2 | 6·25 | 73 | ,, ,, ,, |

As regards the output of ice per h.p. of the 18-ton machine, it was $\frac{100 \text{ h.p.}}{18 \text{ tons}} = 5 \cdot 55$ h.p. per ton of ice made if rated at the maximum horse-power required, but as this horse-power decreased as the ice was being formed, perhaps it would be fairer to take the average horse-power during the 24 hours, which was about 76·5 h.p., making $\frac{76 \cdot 5 \text{ h.p.}}{18 \text{ tons}} = 4 \cdot 25$ h.p. per ton of ice made. With ice plant the load was at a maximum when the brine tank was newly charged with the moulds of fresh water to be frozen, and the minimum load was at the time the ice was formed and ready for lifting. With the plant described, the load was considerably lessened in the evenings and during the night time owing to the condensing water being much lower in temperature than during the heat of the day. To have kept a fairly constant load on the machine during the 24 hours, it would have been necessary to fill and empty a certain number of the total number of moulds in the brine tank periodically, but this procedure was impracticable for the business, because all the ice had to be taken out first thing every morning.

The writer's records of the weight of coal used per ton of ice made were incomplete, but from a boiler test made when the machine was at its maximum load, it was found that they burned 428 lb. of coal per hour, and as the ice made per hour was $\frac{18 \text{ tons}}{24 \text{ hours}} =$ 1,680 lb., this gave only 3·925 lb. of ice per pound of coal. This figure, of course, was only for the ice made during the very hottest part of the day, and actually a greater weight of ice per pound of coal would be made when taking the average weight of coal used per hour during the 24 hours.

All the ice made was "clear" ice for household use, and to prevent the ice from turning out "opaque," the water in the moulds was kept in motion during the time of freezing by a device which gave a pulsating action to the water. This device consisted of a series of pipes laid over the moulds and under the brine-tank covers, to which were attached small cylindrical sheet-iron vessels depending from the pipes and having a small pipe from each of the vessels dipping into the water in the

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moulds. The conical bottom of each vessel had a rubber ball-valve, which closed over the small pipe entering the water, and as the whole system was connected to an air-pump driven by a belt from the brine propeller-shafting, the action was that at each stroke of the pump a small quantity of water was drawn up from the water in each mould and returned again, the ball-valve closing in time to prevent air from being blown into the water. This device gave very good results and took very little power to work it, and when the ice was about ready for lifting, the apparatus was disconnected to prevent the pipes dipping into the moulds from being frozen up. Each block of ice made, and which weighed about 1 cwt., showed just a thin "opaque" core, but the greater bulk of the block was very transparent and hard. This method of obtaining clear ice was of great service when preparing ornamental blocks of various shapes having fruits or flowers frozen within for decorative purposes. The temperature of the brine when all the ice was ready for lifting in the mornings was about $-10°$ C. (14° F.).

The Liverpool Refrigeration Co. Ltd. wrote that they would heartily welcome reliable data on many points connected with refrigerating machinery which up to the present had not been fully investigated. They thought that Dr. Grindley's Paper was of considerable theoretical interest. The idea of cooling the liquefied refrigerant by means of the gas returning to the compressor was not new, and as the gas heated at constant pressure would expand, a larger compressor would be necessary to circulate the same weight of refrigerating fluid. The practical impossibility of isothermal compression by water-jacketing would, they feared, greatly minimize any advantage to be gained. Mr. Anderson's Paper was of more practical interest, and gave much useful information collected, no doubt, from the best available sources. It also emphasized the great lack there was of reliable and well arranged data.

They cordially supported Mr. Anderson's appeal for standardization, and for research, and noted that his suggestions for the latter were of a very practical kind. He did not include

in his list of subjects, the Transfer of Heat through Coil Surfaces at the temperatures usual in refrigeration, and the influence of density, velocity, and temperature difference on the rate of transfer between the refrigerating fluid and the water, brine, or air, etc. They hoped that any data published would be not only in metrical but also in English units, and would be glad to join in any Conference for the discussion of a British standard ton of refrigeration.

Mr. Thomas B. Morley (University of Glasgow) wrote that since there had been proposed, and even actually were in operation, refrigerator cycles other than the usual one described in the Paper, the best standard cycle of comparison would be a more absolute one, and of such the Carnot cycle, which was a "perfect" one, was well known and provided an excellent standard.

*Temperature Limits for Standard Cycle of Comparison.*—The lowest temperature at which the discharge of heat from the refrigerant to the condenser could possibly take place was that of the available supply of condensing water, that is, the temperature of the water *entering* the condenser, $\tau_1$ say. Actually the condensing water was raised in temperature and left the condenser at $\tau_2$ say, a temperature depending chiefly on the abundance of the condensing water. Linde had advocated the adoption of $\tau_2$, as the upper limit of temperature for the cycle of comparison, on the grounds that it was impossible for the temperature at which the condensation of the refrigerant took place to be lower than $\tau_2$, and that if $\tau_1$ were taken as the temperature limit of the ideal cycle, the ratio of the coefficients of performance of the two machines, the actual and the standard (which might be called the "efficiency ratio"), would depend partly on the abundance of the condensing water, as well as on the refrigerating plant itself. While this point was worthy of consideration, there were more weighty reasons why the temperature $\tau_1$ of the *entering* water should be taken for the standard cycle. Many condensers were partly air-cooled as well as water-cooled, so that $\tau_2$ was not entirely controlled by the amount of water used, and by adopting $\tau_2$ for the standard cycle improvements

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in condenser design would not be reflected in the "efficiency ratio." Since this "efficiency ratio" was not the form in which the performance of a plant was stated for commercial purposes, there was no reason why the standard of comparison should not be a stringent one. Perhaps the best plan would be to state two "efficiency ratios" based on both the above temperatures.

Similarly, the lower temperature limit should be that essential to the actual service on which the plant was engaged (ice-making, cold storage, air desiccation, etc.) and not that in the evaporator coils or even in the brine.

"*Undercooling*" *the Liquid.*—While it was true that, for an assigned condensing temperature, cooling the liquid refrigerant before it reached the expansion-valve was advantageous, it would be better to consider this advantage from a different point of view. The temperature to which the liquid could be cooled (with the usual cycle) was $\tau_1$, the inlet condensing water temperature, and rather than condensing at a certain temperature, more or less arbitrarily chosen, and cooling then to $\tau_1$, the desideratum was to keep the condensing temperature itself as near $\tau_1$ as it was possible to do, without unduly retarding the rate of heat transmission in the condenser. It should be noted that the possibility of condensing at $\tau_2$ and then cooling the liquid to $\tau_1$ was another reason for choosing $\tau_1$ rather than $\tau_2$ as the upper temperature of the cycle of comparison.

"*Wet versus Dry*" *Compression.*—Whatever might be the relative merits and demerits of "wet" and "dry" compression, a certain amount of superheat at the end of the compression was desirable. It showed that no liquid was being idly pumped round the whole circuit as liquid, and it ensured that the clearance space at the end of the stroke contained no liquid, which, apart from mechanical dangers, would evaporate during the suction stroke and seriously diminish the volumetric efficiency of the compressor.

*Theoretical Comparison of Refrigerants.*—In Appendix IV of Mr. Anderson's Paper, the first Table on page 1019 gave values of the "unavoidable loss" represented by the area under A F (Fig. 37, page 1017). Even a Carnot refrigerator could not utilize the heat

under A F, so that it never could be available for refrigerating effect at all. Hence the word "loss" in this connection was inappropriate.

The only real indication of the relative value (on purely thermodynamic grounds only) of the refrigerants compared was that shown in the last column of the Table at the foot of page 1020, percentage loss from Carnot's cycle.

*Dr. Grindley's Proposed Cycle.*—The impossibility of approaching isothermal conditions in the compressor had been pointed out by the first speakers in the discussion. He (Mr. Morley) would add, however, that in the tests carried out by Professor Denton * on a large ammonia refrigerator, he found that the compression curve was very nearly adiabatic, even though the compressors were provided with ample water-jackets. Dr. Grindley's cycle had, of course, the advantage that the refrigerant was even from the very beginning of the compression above the temperature of the jacket water, whereas in the usual cycle it was above the jacket temperature during only the latter part of the compression. In spite of this, however, a consideration of the probable heat exchanges between the refrigerant and the compressor walls (which had, unfortunately, to be founded on very meagre experimental evidence) led to the conclusion that Dr. Grindley's cycle would be adversely affected to a much more serious extent than the usual cycle.

*$\psi$-$i$ Diagrams.*—If isothermal compression could not be even approximately realized in practice, the $\psi$-$i$ diagram became useless, but there was a point which called for comment. The term "thermodynamic potential" which Dr. Grindley applied in Fig. 43 (page 1048) to the quantity $\psi$ was the name which, according to Ewing (Mechanical Production of Cold, Appendix F), had been already given by Willard Gibbs to the quantity $i$. Perhaps Dr. Grindley in his reply would give some further explanation of the method of plotting the $\psi$-$i$ diagram, which was not very clear.

*An Improved Cycle for $CO_2$.*—In the discussion of a Paper on $CO_2$ refrigerators published recently by a kindred Institution, an

* Trans. American Society of Mechanical Engineers, 1890, vol. xii, page 326.

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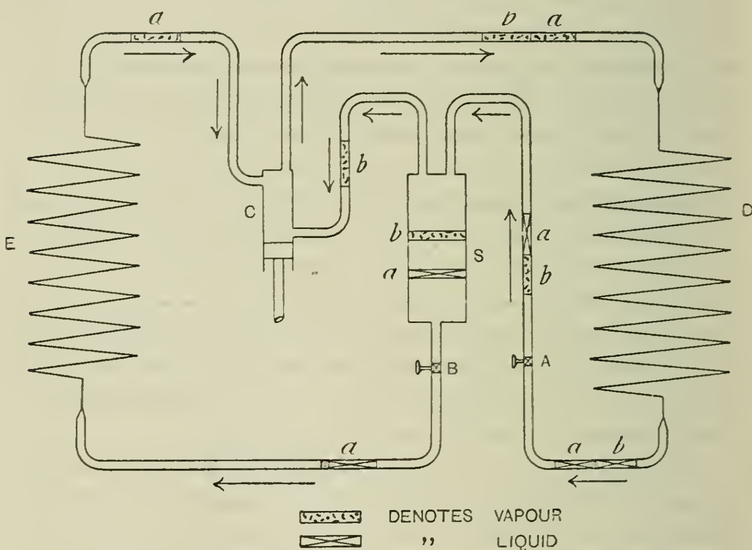
account of an improved cycle was given, which was worthy of a wider audience, and he (Mr. Morley) proposed to describe it. He had added also a method of representing the cycle on the temperature-entropy diagram and of calculating its theoretical possibilities. The plant was shown diagrammatically in Fig. 56. It consisted of the usual compressor C, condenser D and evaporator E, with the addition of an extra expansion-valve, a separator S, and a pipe connection from the separator to the

Fig. 56.—*Improved $CO_2$ Cycle.*

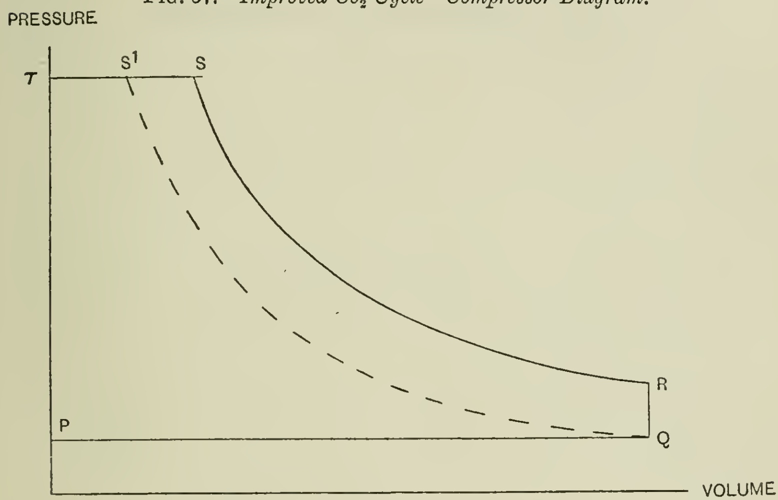



compressor. In the ordinary cycle, the $CO_2$ entering the evaporator-coils through the expansion-valve carried enough energy with it to cause the evaporation of a considerable amount of it, without drawing upon heat taken from the brine. In this new arrangement the expansion took place first of all through the expansion-valve A into the separator S, in which the pressure was higher than that in the evaporator E.

Suppose that of the total amount of refrigerant $a + b$, say, discharged from the compressor and liquefied in the condenser, the

amount $b$ evaporated on passing the first expansion-valve A. In the separator, this vapour $b$ was removed from the remaining liquid $a$. The liquid $a$ then passed through the second expansion-valve B into the evaporator-coils, reaching them with a much smaller proportion of vapour in it than that due to a single-stage expansion. After evaporating, $a$ went to the suction side of the compressor. At the end of the suction stroke, the gas $b$ from the separator was also admitted to the compressor; $a$ and $b$ were then compressed

Fig. 57.—*Improved $Co_2$ Cycle—Compressor Diagram.*

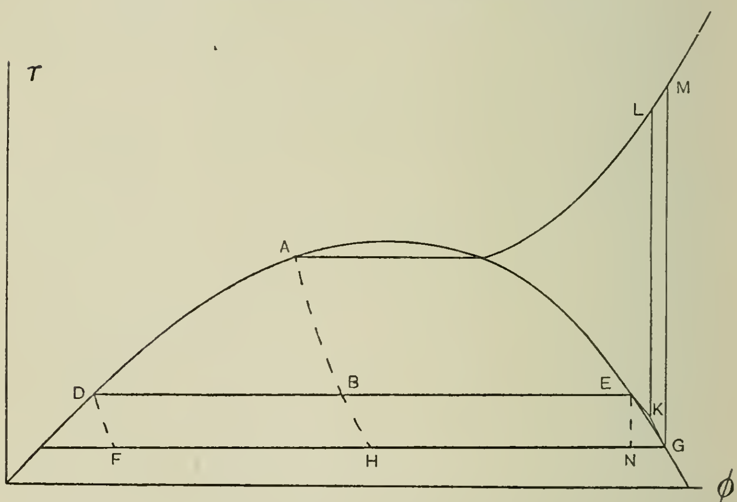



together, discharged and condensed, so completing the cycle. It would be seen that for a given size of compressor, that is, for a given amount $a$ of refrigerant passed through the evaporator, this cycle produced a refrigerating effect due to the evaporation of $a$ from a condition in which it contained very little vapour, whereas on the ordinary cycle the refrigerating effect was only that due to the evaporation of $a$ from a condition in which it might be about half vapour already. The work done in the compressor was increased, since on the compression and discharge stroke it had to deal with $a$ and $b$ together, as compared with $a$ alone in the ordinary case. This was shown in Fig. 57; P Q represented the suction, Q R the

(Mr. Thomas B. Morley.)
rise of pressure due to the admission of $b$, R S the compression, and S T the discharge. For the ordinary cycle the work done was shown by P Q $S^1$ T. The increase in the work done in the compressor was, however, more than counterbalanced by the gain of refrigerating effect.

To represent the cycle on the entropy-temperature diagram, let A, Fig. 58, represent the condition of the liquid leaving the condenser. After passing through the first expansion-valve, its

Fig. 58.—*Entropy Diagram for improved $CO_2$ Cycle.*




condition was represented by B, where B was so chosen that the enthalpy $i_B = i_A$. The dryness-fraction of the substance was now $\frac{DB}{DE}$, or if D B represented the amount of vapour $b$, then B E represented the amount of liquid $a$. We had now to trace the changes of $b$ and $a$ separately on the diagram. The one diagram would suffice, it being remembered that the quantities represented were not now the same, so that areas on the diagram must be multiplied by the values of the quantities concerned, in order to find heat exchanges. We might imagine that the energy of the liquid $(a + b)$ at A above that of liquid at D was all given to $b$, and hence that the area under D E $\times$ $b$, representing the energy

given to $b$ to evaporate it $= i_A - i_D$. E represented the condition of the vapour $b$ after separation. (For the present purpose the fact that a constant-pressure line through A would lie slightly to the left of D might be disregarded.) The liquid $a$ at condition D now passed through the second expansion-valve and reached the evaporating-coils in the condition F. F G represented the evaporation (assuming dry compression was to be followed) of $a$ in the evaporator, and the area under F G $\times a$ represented the heat abstracted from the brine. The quantities $a$ at condition G and $b$ at condition E were now mixed, and the result was $a + b$ at condition K. K was determined from two conditions: first, $i_K \times (a + b) = i_G \times a + i_E \times b$, that is, the total enthalpy values before and after mixing were the same; and, secondly, the pressure at K was that resulting from the mixture of $a$ and $b$ at initially different pressures. The separator was provided with a float-governed outlet, the function of which apparently was to ensure that, when a steady regime was reached, $b$ and $a$ would maintain their proper proportions. It was difficult to predict the position of K from purely thermodynamic considerations, as the pressure in the separator for a steady regime depended on several factors. K L represented the compression, and L A the rejection of heat to the condenser, the heat rejected being represented by the area under L A $\times (a + b)$. The ordinary cycle was A H G M A, and the comparison of the two was shown as follows:—

| —— | Ordinary Cycle. | Improved Cycle. |
|---|---|---|
| Refrigerating effect . | Area under HG $\times a$ $= (i_G - i_H)\,a$ | Area under FG $\times a$ $= (i_G - i_F)\,a = (i_G - i_D)\,a$ |
| Heat to condenser . . | Area under MA $\times a$ $= (i_M - i_A)\,a$ | Area under LA $\times (a + b)$ $= (i_L - i_A)\,(a + b)$ |
| Work spent in compressor . . . | Difference of above $= (i_M - i_G)\,a$ | Difference of above |

While the entropy-temperature diagram showed most clearly, or at least in the most familiar manner, the nature of the operations in the cycles, actual calculations were best made from

(Mr. Thomas B. Morley.)

the enthalpy values $i_L$, $i_G$, etc., which could be obtained from Mollier's $\phi - i$ diagram.

As already mentioned, it was difficult to locate properly the point K, and to assign a value for the pressure in the separator. Also, since the work spent in the cycle was obtained as the difference of two comparatively large quantities, one of which depended on the position of K, the results of calculations from the diagram could only be taken as indicating the effect of the modified cycle in a very general way. Although bearing this in mind, it might still be of interest to compare the results of such calculation with results of actual tests which were quoted in the discussion referred to before another Institution. With condensing water at 30° C., evaporator at $-20°$ C., and *assuming* the intermediate temperature in the separator to be $-10°$ C., the calculation gave the following results (in Continental units from Mollier's diagram):—

| — | Ordinary Cycle. | Improved Cycle. | Percentage of Increase in Improved Cycle. |
|---|---|---|---|
| Refrigerating effect . . . . . . | 14·7 | 30 | 104 |
| Work spent . . . . . . . . . . | 6·6 | 11·9 | 80 |
| Coefficient of performance . . . | 2·23 | 2·52 | 13 |

It should be noted that the separator temperature had a very great influence; for example, if it were 0° C., the calculations gave the increase of output as 84, and increase of the coefficient of performance as 60 per cent.

The figures from an actual plant, with brine at 0° F., 5° range in both evaporator and condenser (the separator temperature was not quoted), were as follows:—

| | | | | |
|---|---|---|---|---|
| Cooling water temperature . °F. | 65 | 75 | 82·5 | 90 |
| Increased output . . . per cent. | 64·2 | 81·3 | 114 | 176 |
| Economy of power per unit of refrigeration . . . . . . . per cent. | 17·4 | 24 | 36 | 50 |

Mr. Bernard Rathmell wrote that, in view of the continuous and rapid increase in the use of refrigerating machinery for a variety of industries, he thought Mr. Anderson's Paper was valuable, especially as its main object appeared to be to urge the necessity and importance of close research work in connection therewith. Admittedly, there were but little practical and reliable data available to the profession generally concerning the relative merits of the different systems in actual working, nor of the relative merits of different types of machinery employed in any system. Manufacturers of machinery had, of course, done a considerable amount of experimenting in the course of their progress, but the outcome of this was not available to the profession generally. It might even be doubted whether such experiments on the average had gone any further than was necessary to establish to a firm's satisfaction the desirability or otherwise of introducing innovations—from a commercial point of view. Such work might be too much bound up with questions of existing patterns, works equipment, and other manufacturing contingencies. It was to be hoped that the Paper and discussion would bring machine manufacturers together, with a view to fixing definite units of rating and performance.

Mr. Anderson, in his remarks on "wet" and "dry" compression, claimed that the "flooded system," so widely used in the United States, was an adaptation of the British method of working, which was moderately "wet." This seemed open to doubt; the flooded system did not appear to lend itself any better to working "wet" than working "dry," intercepters being used to stop any liquid brought over from the coils. The main object of the system was to prevent the gas generated in taking out the liquid heat from passing through the coils, where it could only have the effect of blanketing some of the cooling surface.

In describing and illustrating three double-acting ammonia-compressors (pages 966–70), the author alluded to them as showing the degree of standardization attained. Did they not show too much of this quality? All followed the lines of the compressors introduced by the Linde Co., although all showed variations. All

(Mr. Bernard Rathmell.)

had the defect—and it was a defect, however perfect material and manufacture might be—of valves which projected into the cylinder, and always presented possibilities of accident. All had the disadvantage that a number of joints had to be broken before a cylinder could be opened for inspection; and in marine installations, at any rate, which came under the Registry Societies, these inspections were fairly numerous. However well this type of compressor had performed, there were surely other methods of designing compressors on both efficient and convenient lines.

Figs. 16 and 17 (pages 977 and 979) showed $CO_2$ compressors, the salient point of each being that it had a "liner fitting for practically the whole length." Why was a liner fitted at all, if it did not cover the whole of the piston travel? In the examples shown, the piston-rings, presumably, did not travel beyond the joint of the liner; this involved some peripheral clearance at each end, and a longer piston than was really necessary. There was probably no difficulty in turning these out a good job in the first place, for the whole could be bolted together for the finishing off; but what was the position if it became necessary to renew a liner, say, on shipboard in mid-voyage? Judging by the compressors mentioned and by other examples of current practice illustrated, it could hardly be claimed that manufacturers had reached finality.

The Tables and Figures given in the Appendixes for $CO_2$, $NH_3$, and $SO_2$ gases were of great interest and value, as embodying the latest and corrected figures on these points.

Mr. WILFRID STOKES wrote that he had read with much interest the Paper by Dr. Grindley, more particularly as it drew attention to the very considerable opening which existed for the improvement in $CO_2$ machines, to which end the writer had been conducting a very extensive set of experiments working in accordance with the system invented by Mr. G. T. Voorhees. In Dr. Grindley's new cycle, various practical difficulties presented themselves which had been referred to in the discussion. These difficulties appeared to him to be so great as to render the cycle impracticable.

As was well known, a very considerable amount of liquid $CO_2$ became gas on passing the expansion-valve in order to cool the $CO_2$ from the temperature of the condenser to that of the evaporator. Dr. Grindley proposed to reduce this waste by cooling the $CO_2$ after leaving the condenser, by the gas coming from the evaporator. In the Voorhees system no attempt was made to reduce the amount of gas produced in cooling the $CO_2$ liquid, but, by the introduction of an automatic second expansion-valve and a separator vessel, the majority of the gas was short-circuited so as not to pass through the evaporator at all. By the use of suitable valves, this gas was admitted into the compressor at the end of its suction-stroke at a higher pressure and temperature than the evaporator suction gas, and thus considerable economies in power were obtained, as well as a materially increased output from the compressor.

In a small machine he (Mr. Stokes) had used for his investigations, he had obtained the following results without any more difficulty in working than with a normal machine:—

| Temperature of Cooling Water. | Increased Output from same Compressor. | Economy in Power per unit of Refrigeration. |
|---|---|---|
| ° F. | Per cent. | Per cent. |
| 65 | 64·2 | 17·4 |
| 75 | 81·3 | 24·0 |
| 82·5 | 114·0 | 36·0 |
| 90 | 176·0 | 50·0 |

These results were with wet compression and with 0° brine and 5° range in both evaporator and condenser, and with the machine running at the same speed and under similar conditions in each comparative test. It would be observed that the saving in power was practically the same as that given by Dr. Grindley in the Table (page 1037) for brine at + 5° brine, that is, 0° evaporator, when working under his proposed new cycle.

(Mr. Wilfrid Stokes.)

The principal reasons why so large an economy could so easily be obtained with the Voorhees system were as follows, taking 90° cooling water as the hottest to be considered :—

(1) A large proportion of the gas formed after the expansion-valve never had to be reduced to the evaporator temperature (up to 50 per cent.).

(2) The gas from the evaporator was partially compressed without mechanical effort by the superimposed charge of gas from the separator vessel (up to double the evaporator pressure).

(3) This partial compression took place without heating the gas, and its temperature was therefore lower in the compressor (up to 30° F.).

(4) As compressors fitted with leathers were run with "wet" compression, the amount of liquid taken from the evaporator and wasted was reduced for reason No. 3 (up to 8 per cent.).

(5) As the clearance and leakage losses were constant, they formed a lower percentage of the increased output (up to 60 per cent. reduction).

(6) As the mechanical friction losses both in prime mover and compressor were practically constant, they formed a lower percentage of the increased output (up to 60 per cent. reduction).

Although in the small machine hitherto experimented with, certain of the losses were not normal, it was interesting to note that the gains obtained in practice had been predicted almost exactly beforehand by the use of Mollier's diagrams referred to by the author.

A machine on a commercial scale was under construction, from which he was confident that results would be obtained in no way inferior to those referred to above. In this machine all the valves were automatic, and the separator vessel contained the second expansion-valve with its automatic control which had been specially designed for ship use, as it was not affected by the motion of the vessel. With regard to the question of first cost, the smaller prime-mover required to produce the same refrigeration rendered the total cost of the outfit about the same as that of a normal machine. The very marked economy in working cost was therefore obtained

without any additional first cost of manufacture, and without any increased difficulty in working or keeping in order.

Mr. Hal Williams wrote that he felt that the Institution owed a large measure of thanks to Mr. Anderson for his Paper; only those who had ever undertaken similar work could appreciate what an enormous amount of personal time and trouble it had involved, and for this reason he felt that some of the criticism which was based more on the author's wording than on the obvious meaning of what had been written, had been unduly harsh and misplaced.

The writer had been one of the British Official Delegates to the premier Congress du Froid in Paris in 1908, where the question of the standardization of Refrigeration Machines had been discussed at considerable length. It was discussed again at the second Congress in Vienna in 1910, by the Association Française du Froid at Toulouse this year, and it was to be further discussed at the third Congress at Chicago next September. The whole question bristled with difficulties, chiefly because the use to which a refrigerating machine was put was by no means a standard, and consequently, the design of the complete plant of which the compressor formed only a part, varied in nearly every case; again, some of the standards that had been proposed for compression machines were not applicable to absorption machines, which were again coming into prominence, and vice versa. The size of the compressor had been suggested, but this unit could not be adopted because the amount of gas which it was capable of displacing varied with the back pressures, and this varied with the use to which the plant was being put.

A standard of the nature proposed was designed principally to be of use to the non-technical public, but as refrigerating plants were nearly always bought as a whole, for the purpose of producing certain results under certain conditions which had to be very fully laid down before the plant was designed, the statement or guarantee that a machine was standard under one set of conditions would be no real guide to a purchaser who wanted it to work under entirely different conditions. To be of any real benefit to the purchaser, the standardization would have to be carried through every size of

(Mr. Hal Williams.)
machine for every known application, and would have to apply not only to the compressor in a compression plant, and to the absorber, generator and rectifier in an absorption plant, but to the condenser, refrigerator, and to the pipe surface in a brine or direct-expansion system. It could not even end there, but would have to be carried to the insulation used.

The Association Française du Froid were recommending the following thermal standards :—" The normal power of a freezing machine is the number of frigories it is capable of absorbing in one hour for temperatures of gas, of $+ 25°$ C at the condenser, $+ 15°$ C. at the expansion regulator, and $- 10°$ C. at the refrigerator."

He quite thought that a standard ton refrigeration in B.Th.U. per hour should be established, though most people now worked to approximately the same figures, and that a standard for B.Th.U. absorbed under stated conditions of forward and back pressures, such as that suggested by the French Association, would be useful; when once this had been determined, it would always be possible to figure back to this standard from other conditions. There was no real difficulty in the matter to those who understood the subject, and others should not attempt to deal with matters they did not understand. There was no doubt that widely different views were held by makers of refrigerating plant as to the displacement and surfaces to be allowed for a given duty, but that was more a commercial than a technical matter, and the same thing was found in almost every branch of engineering. Take, for instance, the wide variation in the design of electric motors and dynamos of given horse-powers at given voltages and speeds. By a careful analysis of displacements, speeds and surfaces, a refrigerating engineer had no difficulty at all in deciding the relative values of different tenders, and with the exceptions he had referred to, there was no need in his opinion to alter the existing condition of things.

He had been much interested in the temperature-entropy diagrams that both authors had prepared, and in connection with Dr. Grindley's Paper he desired to point out the labour of transferring to British units was undertaken by him (Mr. Williams) in conjunction with Professor A. W. Porter, of University College, in a book,

"Mechanical Refrigeration," published ten years ago. He readily recognized that Dr. Grindley had gone further, and had dealt with Dr. Mollier's later diagrams, but he thought that some slight recognition should have been made of Professor Porter's and his own pioneer work on the same lines. With regard to the general examination of the theory of the performance of refrigerating machines in Mr. Anderson's Paper, he ventured to claim that no more exact method of treating the question had ever been put forward than in the short outline given in the book referred to. Great care was taken to deal with the irreversible changes involved, especially in regard to the failure of the *p. v.* diagram to represent the external work done in such irreversible changes, and the failure of the temperature-entropy diagram to represent the corresponding heat entry.

Mr. Anderson wrote that, in reading over, first the discussion as contributed by actual speakers with the view of answering those points which time had prevented him dealing with, he noted that Mr. Hodsdon had said "there was nothing for refrigerating engineers in the Papers." He did not wish to discuss the scheme on which his Paper was built, but, broadly speaking, it was intended to bring the Proceedings of the Institution up to date with respect to current practice in refrigeration, and also to outline the theory as far as it was generally known. Incidentally, this would give the "ordinary engineer" an opportunity of taking part in the discussion. He was bound to say that the scheme had worked remarkably well, inasmuch as some of the most important contributions had come from those who would hardly claim to be refrigerating engineers. Further, the note on page 949 gave the real use of the Paper, that it was written "in order that views may be obtained which may be of assistance in the prosecution of the suggested research." Mr. Hodsdon said (page 1089) "he did not think there were many refrigerating engineers who knew exactly what the specific heat of brine was." The author's opinion on reading the discussion most carefully and remembering the petition that led the U.S.A. Government recently to vote $15,000

(Mr. J. Wemyss Anderson.)

for investigations in the subject was, that refrigerating engineers had very little exact knowledge of anything at all in connection with their profession, and this indicated to the Institution the vast field for research that had been opened up.

He (Mr. Anderson) wished to pay tribute to the great value of Dr. Grindley's Paper, as it was an outstanding contribution to the theory of vapour-compression machines. The Grindley cycle might not at the moment appear practicable, with our present knowledge of metals and design, but it gave those interested in the theory, new knowledge and new ideas and those interested in practice, a new ideal, and had called from Mr. Hill (page 1113) a suggestion of great value. It was certain that Dr. Grindley did not in the least merit the criticism of Mr. Hodsdon when he said: "the inference contained in the Paper (Dr. Grindley's) was that refrigerating engineers did not know their business." He greatly regretted Mr. Hodsdon's excellent contribution of exactly the type required was marred by the expression to which he (Mr. Anderson) had taken so strong an exception. Moreover, nothing had been further from the author's mind than to say "that no improvement had taken place as a result of scientific investigation." In his spoken reply the author had transferred the word "published," used on page 950, so that the sentence read "from published research or from accounts of practical progress, than that of mechanical refrigeration." He had used the word "research" in the first instance in the academical sense that research of any value would always be published. He would strongly urge refrigerating engineers to adopt the practice followed by other branches of the engineering profession, and make known the results of research and investigation. Nothing was gained by keeping on the cloak of reserve which had only partially been removed by the discussion on the two Papers. As Captain Sankey had rightly remarked (page 1071), a mere recital of what had been done, without giving results, was useless.

Mr. Brier had called attention (page 1095) to the author's use of the words "attemperated brine." As Mr. Brier had correctly said, "warm" or chilling brine had been used for many years,

but the object of the attemperator (which gave rise to the new expression "attemperated" brine) and its auxiliary attemperator-cock was, as set out on page 996, threefold, but in view of Mr. Brier's remarks, it would perhaps be better to supplement the description of the three types thus: (1) all freezing brine at 5° (say) through the leads, (2) all chilling brine at 35° to 38° (or even above) as regulated by the main attemperator, (3) freezing or chilling brines as required at any temperatures between 5° and 38° (say) as regulated by the auxiliary attemperator-cocks. The temperatures taken were only used as typical examples.

In further answer to Mr. Brier, he would say a $CO_2$ machine was often preferred on a large passenger steamer, because of the entire absence of smell. $NH_3$ machines were not at present in use on warships because of the probable effect in an engagement with the enemy, as a pipe might burst with an appalling effect on the crew. In his own opinion, a captain who went into action with his refrigerating machine charged—either $CO_2$ or $NH_3$—committed a grave error of judgment—with $NH_3$ for the reasons just given, and $CO_2$ because of the risk of asphyxiating the men below decks.

The list of details given for testing refrigerating machines were absolutely essential, if true comparisons were to be made. It was necessary to repeat that the details usually given made it quite possible for an expert to allow any machine he wished to come out as the best.

He fully appreciated Mr. Enock's remarks (page 1096) and the distinctive features of his firm's compressor designs. Volumetric efficiency was a most important subject, but it would appear that a great amount of fundamental work had yet to be covered before research or discussion on that particular section would be really profitable. A better knowledge of the physical constants and properties of the refrigerants was certainly required.

With respect to the Communications, he would say in answer to Mr. Beck (page 1109) that the condenser shown on page 963 was not a double-pipe condenser. It was a type used extensively in the United States of America, but like the double-pipe condenser shown on page 965, it did not brook neglect. The compressor displacement

(Mr. J. Wemyss Anderson.)

given on page 1021 was quite correct for a *mean* condenser temperature of 70° F., but he had no hesitation in saying that even with condensing water *on* at 80° F. that the No. 5 compressor quoted by Mr. Beck had something seriously wrong with its design—probably excessive clearance, provided that the rest of the plant (coil surface, etc.) was correctly designed.

He strongly commended Mr. Buckton's remarks (page 1110) to refrigerating machine makers; they were full of pregnant suggestion, and outlined probable practical developments.

Mr. Douglas (page 1112), The Liverpool Refrigeration Co. (page 1121), and Mr. Hodsdon (page 1092), had suggested, as a possible research, the transmission of heat through coil surfaces at temperatures used in refrigerating machines. He cordially supported this suggestion, and had mentioned at the top of page 1025, that the ordinary laws for the transmission of heat through metal walls did not hold good for low temperatures and small temperature differences. He was certainly pleased to find such important firms as Messrs. Douglas and The Liverpool Refrigeration Co. supported his contention that more reliable data of all kinds were required in refrigeration, and that they would support any movement towards standardizing the ton of refrigeration.

He thanked Mr. Hill (page 1113) for his kindly appreciation, and thought that in connection with Dr. Grindley's cycle, Mr. Hill had made the only practical and undoubtedly valuable suggestion with respect to how the cycle could, perhaps, be approximately carried out, namely, by using the De La Vergne system of oil injection. With an efficient oil-cooling arrangement, the author was of the opinion that good results would be obtained.

Mr. Knox (page 1114) had sent a most interesting communication. Personally, without detailed information, he could not understand why the larger plants were relatively more inefficient, but he could not accept the suggestion that relatively efficient machines in this country became relatively inefficient in the tropics.

Mr. Morley (page 1123) proposed an ideal machine working on the Carnot cycle as a standard machine of comparison. He disagreed entirely with this suggestion. The reciprocating steam-engine, the

gas- and oil-engine (constant pressure), the gas- and oil-engine (constant volume), had each its own standard of comparison which was certainly not the Carnot cycle, and he submitted that a standard was required for each distinctive type of refrigerating machine, of which the compression and absorption systems were of outstanding importance. He thought Mr. Morley quibbled over the expression "unavoidable loss." Loss or no loss, the Carnot cycle penalized this quantity very severely. Both Mr. Morley and Mr. Stokes had added to the value of both Papers by expounding Mr. Voorhees' new cycle for $CO_2$ machines.

He thanked Mr. Hal Williams (page 1133) for his contribution. He was glad that such an authority, both theoretically and practically, as Mr. Williams, had used the expression "obvious meaning," because he (Mr. Anderson) had to admit that what was intended as a gentle reminder that much more interchange of opinions, of ideas and of actual results (research and practical) were required in the refrigerating world, had raised a veritable hornets' nest. Mr. Rathmell (page 1129), who had also sent a valuable communication, had, he thought, with respect to this point, summed up the situation rather well when he wrote "it might even be doubted whether such (manufacturer's) experiments on the average, had gone any further than was necessary to establish to a firm's satisfaction, the desirability or otherwise of introducing innovations."

In conclusion, he would say that the discussion clearly indicated that a vast amount of research (published) was needed to bring the science of refrigeration up to the standard that obtained in other branches of engineering, and he thought that the Table given on page 1000, might as a result of the discussion be extended, by

(*g*) The physical constants and properties of the chief refrigerants.

(*h*) The transmission of heat through coils under the varying conditions found in refrigerating practice.

(*i*) The specific heats and specific densities of calcium chloride brine at the varying temperatures and densities used in refrigeration, and also the best means of measuring the quantity of brine circulated in the machine.

(Mr. J. Wemyss Anderson.)

(*j*) The volumetric efficiencies of compressors with particular reference to (I) the refrigerant used, (II) the amount of superheating under varying conditions, (III) the relation of diameter to stroke, (IV) the amount of clearance, and (V) the velocities of the gas with respect to the valve openings.

He wished to thank the President and members of the Institution for their very cordial reception of his Paper. He appreciated the healthy and vigorous discussion which had followed the reading of the two Papers, and thought there could be no doubt that interest in the subject had been greatly stimulated.

Dr. Grindley wrote, thanking the various contributors for their very welcome criticism on his Paper. Referring to the early part of the Paper, when a new cycle of operations was described which would lead to increased performances of vapour-compression machines, the majority of the critics had assumed that he had put forward isothermal compression as a practical proposition by water-jacketing the cylinder only. Such was not the case. The author's intention was to put forward the claims of a new and simple theoretical cycle, which offered advantages over the theoretical cycle to which present practice tended. The perfect attainment of this cycle under the ordinary practical limitations was clearly impossible, but, even in an imperfect form, this cycle offered great advantages over the old one, and it was quite possible that this could be obtained by the water-jacket. The general assumption made by those who had discussed the new cycle was that the only alternative to isothermal compression was adiabatic compression, but the discussion had brought to light some evidence which showed that even with the present cycle, where the mean temperature of the vapour in the compressor was not much above that of the jacket-water, an interchange of heat did occur, so that a much greater interchange of heat should occur when the vapour in the compressor was at a temperature always higher than that of the jacket-water. The compression would then be quite different from the adiabatic assumed, and any heat whatever extracted from the vapour in the compressor meant increased performance of the

machine, for only with perfect adiabatic compression combined with the use of the interchanger would the theoretical performance fall to that of the old cycle of operations.

A second method of approximating to the cycle would be with multiple-stage compression combined with intercoolers, as described by Captain Sankey (page 1068); and if Mr. Buckton's remark (page 1111) that "it remained to be seen if a multiple-cylinder, high-revolution type of geared machine would oust the present partially standardized horizontal, long stroke, belt-driven machine from its present position" should truly foreshadow developments, then the proposed cycle would be the one aimed at, and the increases in performance described in the Paper largely realized in practice.

Professor Jenkin had given a most valuable criticism of the proposed cycle (page 1055), but his remark that the interchanger was of no use at all, unless at the same time one could do isothermal cooling, hardly covered the facts for the reasons stated above. Everybody agreed that isothermal compression was an ideal one only imperfectly obtained in practice, but any deviation whatever from the adiabatic towards the isothermal in the new cycle meant a gain in performance. He (Dr. Grindley) was glad to read Professor Jenkin's note that, taking the comparison of compressor volumes per unit of refrigeration, the new cycle worked out best. Like Professor Jenkin, he would be glad to see some accurate experiments made to determine the heat properties of refrigerants, and he waited with much interest for the publication of Professor Jenkin's work in this direction.

The proposed cycle had received severe comment from Mr. Voorhees, who, along with Mr. Morley and Mr. Stokes, appeared to be pressing the claims of a further new cycle of operations. Mr. Voorhees said (page 1059) that adiabatic compression was the only alternative to isothermal cooling, and in his calculations on the cycle proposed by the author, besides this assumption of adiabatic compression, he unnecessarily increased the power required by 18 per cent. With other assumptions to which he referred as taking the figures in the fairest way, he concluded that a 20 per cent. loss of performance would result. His calculations on the cycle were incorrect, since—

(Dr. J. H. Grindley.)
(1) the adiabatic was not the only alternative to the isothermal, and (2) the power and refrigerating effect per pound were given directly by the $\phi$–$i$ diagram, and the power therefore did not need the 18 per cent. increase given by Mr. Voorhees.

Mr. Morley (page 1125), who detailed the second new cycle referred to above, showed a $\tau$–$\phi$ diagram for the cycle in which the work done was given as a difference of two comparatively large quantities, one of which depended on the problematical position of a point K on the diagram. An increase of performance was shown of 13 per cent., but in saying that this gain in performance could be increased to 60 per cent. if the temperature of the interchanger were 0° C., Mr. Morley must be in considerable error, for his (Dr. Grindley's) calculations showed that it was the increase in power, not performance, which went up 60 per cent., the increase in performance being not more than 16 per cent.

Before commenting on the figures given in the Table on page 1128, which were repeated by Mr. Stokes on page 1131, it would be necessary to know full details of the experiments and method of comparison, for such large increases as shown in the Table were much greater than those shown to be probable by the $\tau$–$\phi$ diagram. The true test as to the value of a new cycle of operations was to compare the performance of machines specially designed to obtain the best out of each cycle.

Among Mr. Willcox's very interesting remarks (page 1062) were found references to the subject of compound compression and stage compression with intercooling. He (Dr. Grindley) could only express regret that Mr. Willcox gave no information as to the result of experimental work on these matters, for it must be obvious to him that, as Capain Sankey had remarked (page 1071), a mere recital of work done without giving results was not of much use in discussion.

Captain Sankey had shown in his valuable contribution (page 1068) how stage compression with intercooling would realize much of the gain in performance shown by the proposed cycle. He (Dr. Grindley) had worked out the case of a simple two-stage $CO_2$ compressor with intercooler and working with the interchanger.

With temperature limits 10° F. and 88·5° F., as in the Table (page 1037), the gain in performance would be at least 18 per cent. over the old cycle, and with temperature limits 10° F. and 90° F. the increase in performance would be at least 27 per cent. With three-stage compression, these gains would be increased. The cycle proposed, however, was the limit to which the above stage compression tended when combined with intercooling between cylinders and undercooling by means of the interchanger, and he (Dr. Grindley) thanked Captain Sankey both for his criticism of the new cycle and for his advocacy for the use of the $\tau$–$\phi$ diagram, which was not always clearly understood and its advantages realized.

He (Dr. Grindley) quite agreed with Mr. Harrap (page 1071) that to gain economy in one part of a cycle and lose it in another was of no advantage, but he (Dr. Grindley) would point out that this was not what was proposed in the cycle he described. The precooling, which Mr. Harrap said was not used nowadays, had not been accomplished in the manner described in the Paper.

The alleged unfortunate miscalculation by him (Dr. Grindley) of the great increase of power in the new cycle, as mentioned by Mr. Tyler (page 1079), was surely a more unfortunate miscalculation on Mr. Tyler's part. After the power expended and the refrigerating effect produced per pound of refrigerant had been calculated correctly from the diagram, Mr. Tyler proposed to increase this power by 93 per cent., making $W_2 = 34{\cdot}5$ instead of $17{\cdot}9$, and showing a resulting loss in performance of 26 per cent. for the new cycle. This increase of power by 93 per cent. was of course incorrect, for the $\phi$–$i$ diagram gave both the power and the refrigerating effect per pound correctly.

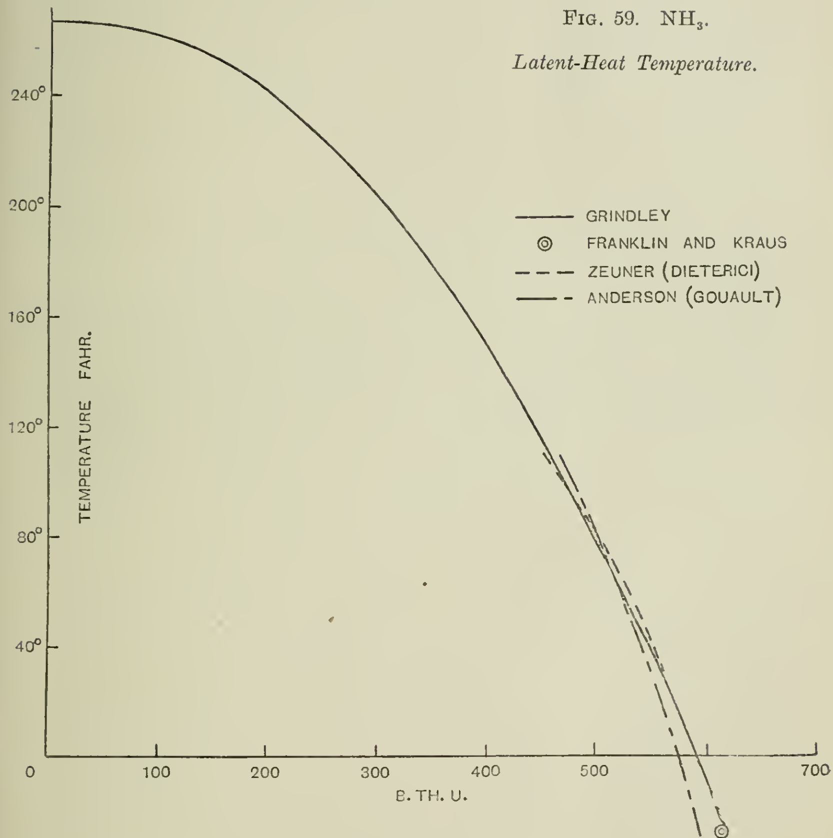
Mr. Hodsdon (page 1081) raised the question of the value of water-jackets on $CO_2$ machines, and his remarks thereon have been of great interest, but he (Dr. Grindley) would like to point out that the experience with dry-compression machines that the water frequently left the jackets cooler than it went in, was shown by the $\tau$–$\phi$ diagram to be both possible and probable, since the mean temperature inside the compressor cylinder might be lower than

(Dr. J. H. Grindley.)
that of the jacket-water. The practical higher limit of temperature in the cylinder was given by Mr. Hodsdon as 150° F. or 160° F., and he said that under such conditions the interchanger would be of no use. Even if this temperature limit could not be attained by a properly designed water-jacketed compressor, a two-stage compressor with intercooler could easily work below this temperature limit, so that Mr. Hodsdon's sweeping condemnation was unsound.

The $\tau - \phi$ diagram for $NH_3$ shown by Mr. Hodsdon, when compared with that given on page 1046, showed the advantage of using skew co-ordinates when preparing such diagrams. With regard to the Table for the properties of $NH_3$ given on page 1051, Mr. Hodsdon had made some valuable remarks. He gave on Fig. 49 (page 1084) a number of lines showing the connection between the latent heat and the temperature as given by various authorities, and said that he (Dr. Grindley) was not fair in going outside at the right-hand bottom corner. He (Dr. Grindley) would point out that, so far as he was aware, most of these curves, for temperatures below 32° F. were based on slight experimental evidence, and it was his opinion when preparing the Table that the best experimental result on the value of L at any temperature below 32° F. was that of Franklin and Kraus, who found that at ordinary atmospheric pressure the latent heat of $NH_3$ was 614 B.Th.U., and the temperature of evaporation $-$ 27·4° F. This value of L was in fair agreement with the calculated result of van 't Hoff. Combined with this result, he (Dr. Grindley) had the results of Dieterici's experiments above 32° F., and the further data that the critical temperature was 266·9° F., the latent heat being then zero. The curve which best represented these results was given by the equation (1), page 1039, which when plotted over a range of temperature from $-$ 40° F. to the critical temperature gave the curve shown on Fig. 59, on which the results of Dieterici experiments (given by Zeuner) and the figures given by Anderson were also plotted. He could not think it probable, though of course possible, that the true curve would show a fairly sharp deviation to the left, as would be required if Mr. Hodsdon's suggestions were carried out. The large curvatures shown by the

Fig. 59. $NH_3$.

*Latent-Heat Temperature.*



(Dr. J. H. Grindley.)
lines marked Mollier and Ledoux on Fig. 49 (page 1084) also did not appear to him (Dr. Grindley) as probable, so that he was left with the data mentioned above as the best from which to construct the Table given in the Paper.

Mr. Enock's remark (page 1098) that both vertical $NH_3$ and vertical $CO_2$ machines were sometimes made with water-jackets, and operated successfully, showed that there was some merit in the water-jacket even when working on the usual cycle.

With regard to Mr. Knox's experience with refrigerating machines in the tropics (page 1114), he (Dr. Grindley) would like to say that it was with the object of making the working of $CO_2$ machines in the tropics more certain and efficient that he originally thought out the proposed cycle, for it was obvious that, with condensing water at the temperatures described by Mr. Knox, the advantage of using the new cycle, even in an imperfect form, must be very great, greater even than that shown in the Paper.

With some of Mr. Morley's remarks he had already dealt, but with regard to the symbols $i$ and $\psi$ he (Dr. Grindley) would point out that the term "thermodynamic potential" was a general term applied usually to three functions, and one of these was the enthalpy of the Paper, which was identical with the "total heat function" of Clausius, the latter title being a really good one, as the definition of $i$ would show. Another of these potentials was the function $\psi$ of the Paper, which Duheim called the "potential at constant volume." The third of these functions they were not concerned with. Since the Paper dealt with both $i$ and $\psi$, he (Dr. Grindley) only referred to the latter as the potential. To construct the $\psi - i$ diagram, since for any particular condition of a substance we had $\psi = i - \tau\phi$, values of $\psi$ could be calculated at once from the $\phi - i$ diagram, for the latter gave simultaneous values of $\phi$ and $i$ for any temperature $\tau$ in either the liquid, dry saturated or superheated conditions of the substance, and these values of $\psi$ could then be plotted against $i$. Rectangular co-ordinates were used, and the scale of $\psi$ was greater than the scale of $i$, an obvious advantage since W (the power) was less than the refrigerating effect R.

As a useful check on the calculations, it was easy to see that $\psi$ had the same value for the liquid as for the dry saturated vapour at the same temperature and pressure, thus in the liquid

$$\psi_w = i_w - \tau\phi_w$$

and in the dry saturated vapour

$$\psi = i - \tau\phi$$

hence $$\psi - \psi_w = i - i_w - \tau(\phi - \phi_w)$$

but $$\begin{aligned} i - i_w &= \mathrm{H} + \mathrm{A}ps_o - (\mathrm{S} + \mathrm{A}ps_o) \\ &= \mathrm{H} - \mathrm{S} \\ &= \mathrm{L} \end{aligned}$$

and $$\tau(\phi - \phi_w) = \tau\phi_s = \tau\frac{\mathrm{L}}{\tau} = \mathrm{L}$$

so that $$\psi - \psi_w = \mathrm{L} - \mathrm{L} = 0$$

or $$\psi = \psi_w$$

In conclusion, he wished again to thank those members who took part in the discussion for their valuable criticism and kind remarks, and particularly to thank his old colleague, Mr. J. Wemyss Anderson, whose advice in refrigeration matters had always been freely given to him.

# The Institution of Mechanical Engineers.

## PROCEEDINGS.

### DECEMBER 1912.

An ORDINARY GENERAL MEETING was held at the Institution on Friday, 20th December 1912, at Eight o'clock p.m.; EDWARD B. ELLINGTON, Esq., President, in the Chair.

The Minutes of the previous Meeting were read and confirmed.

The PRESIDENT announced that, to fill the vacancy among the Members of Council, caused by the decease of Mr. Henry Lea, the Council had appointed SIR GERARD A. MUNTZ, BART., as a Member of Council. He would retire at the next Annual General Meeting, in accordance with Article 25.

The PRESIDENT announced that the Ballot Lists for the election of New Members had been opened by a Committee appointed by the Council, and that the following ninety-four candidates were found to be duly elected:—

MEMBERS.

| | |
|---|---|
| ADAMSON, DAVID, | Newcastle-on-Tyne. |
| BRAMWELL, BALFOUR, | Belfast. |
| BUCK, GEORGE FREDERICK, | Manchester. |
| CARTER, GEORGE JOHN, | Birkenhead. |
| CHRISTIE, JOHN MURRAY, | Calcutta. |
| ELSTOB, FRANCIS, | London. |

| | |
|---|---|
| Forster, Alfred Lindsay, | Birmingham. |
| Gilbert, Ernest, | Moscow. |
| Griffiths, Ernest, | Liverpool. |
| Higginbotham, Walter, | Glasgow. |
| Hillman, Cecil Robert, | São Paulo. |
| Kaye, Louis James, | Paris. |
| Martin-Davey, William, | Liverpool. |
| Nevill, Walter Elphinstone, | London. |
| Nixon, Willoughby Francis, | Buenos Aires. |
| North, John Frederick, | Pará. |
| Page, Frederick James, | Bombay. |
| Reavell, James Arthur, | London. |
| Townhill, William, | Hull. |
| Tweedie, Frank Forbes, | Negapatam. |

ASSOCIATE MEMBERS.

| | |
|---|---|
| Aitken, James, | Aberdeen. |
| Bird, Harold Hughes, | Derby. |
| Bolton, Edward John, | Stoke-on-Trent. |
| Brand, William Deane, | London. |
| Broughton, Arthur Charles, | Berbice, B.G. |
| Chadwick, Robert Joseph Montague, | Hawke's Bay, N.Z. |
| Conyngham, Ernest Knox, | Jesselton, B.N.B. |
| Dale, John Stanley, | Preston. |
| Daniels, Wilfrid, | Leigh, Lancs. |
| Davies, James Samuel, | Rhondda, Glam. |
| Dickson, Robert Cochran, | Sumatra. |
| Digby, Clifford, | Manchester. |
| Ellis, Robert Battiscombe Askquith, | Birmingham. |
| Golder, James, | London. |
| Gosford, Albert Thomas, | Bombay. |
| Graham, James, | Selangor. |
| Hardwicke, John Augustus Vigor, | São Paulo. |
| Harris, Albert Edward, | Rochdale. |
| Harris, William Augustus, | Trinidad. |
| Hawkins, Burton Tyas, | St. Helens. |

| | |
|---|---|
| Hennessy, Charles Percival, . . | Machen, Mon. |
| Hill, Joseph, . . . . . | Sheerness. |
| Hodson, Reginald Cecil, . . . | London. |
| Hulme, William, . . . . . | Tokyo. |
| Jarvie, William Muirhead, . . . | Singapore. |
| Lawton, Ralph Waldo, . . . . | Calcutta. |
| Leslie-Bredée, Giacomo, . . . | Calcutta. |
| Malan, Lionel de Mérindol, . . . | London. |
| Manley, Edward Lovell, . . . | Saidpur. |
| Mitchell, William Anderson, . . | Titaghur. |
| Murray, John, . . . . . | Tongaat, Natal. |
| Neilson, George, . . . . . | Blyth. |
| Pull, Ernest, R.N.R., . . . . | London. |
| Robertson, George Wallace, . - . | Preston. |
| Suggate, Claude Francis Denny, . . | London. |
| Symons, Angus Bryant, . . . . | London. |
| Taylor, William Arthur Trevor, . . | London. |
| Thomas, Frank George, . . . . | Westland, N.Z. |
| Tuckett, William Fothergill, . . | London. |
| Utley, Reginald, . . . . . | Lytham. |
| Wadia, Ardeshir Dosâbhai, . . . | Ahmedabad. |
| Walker, James Ernest, . . . . | Calcutta. |
| Wallis, Thomas Alexander, . . . | Manchester. |
| Webber, James Trery, Eng. Lieut. R.N., . | Portsmouth. |
| Whalley, Hermes de, . . . . | London. |
| Wheatley, Leonard James Theodor, . | Woolwich. |
| Young, Algernon Gordon, . . . | London. |
| Young, Douglas Stewart, . . . | London. |

ASSOCIATE.

| | |
|---|---|
| White, Carter, . . . . . | London. |

GRADUATES.

| | |
|---|---|
| Ayres, George Herbert, . . . . | London. |
| Bansall, John Wilson, . . . . | Cambridge. |
| Brierley, Walter, . . . . | Manchester. |

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| CARR, REGINALD SIDNEY, . . . . | Doncaster. |
| CHANDLER, WILLIAM RICHARD POWELL, . | Gloucester. |
| DICKSEE, CEDRIC BERNARD, . . . | Wembley. |
| DRUMMOND, WALTER JAMES, . . . | London. |
| GILBERT, REGINALD WILLIAM ARTHUR, . | Ipswich. |
| GOMME, DAVID ESMOND, . . . . | Cachar. |
| JACKSON, RALPH, . . . . . | Manchester. |
| LAWTON, FREDERICK WILLIAM, . . . | Stoke-on-Trent. |
| MANGNALL, ARTHUR RIVINGTON, . . | Bolton. |
| MARRIOTT, JOHN FRANCIS LAYCOCK, . . | Gloucester. |
| MENEZES, JOSEPH ALOYSIUS, . . . | London. |
| ORMROD, ALFRED SMITHELLS, . . . | Horwich. |
| READ, ALFRED HOWARD, . . . . | Liverpool. |
| REEVES, BERNARD JACK, . . . . | London. |
| ROBINSON, ISAAC VINCENT, . . . | Widnes. |
| ROGERS, HORACE EDWIN, . . . . | London. |
| SENEVIRATNE, GANNAGODA DON IRVIN, . | Paisley. |
| SMYTH, SIDNEY, . . . . . | Ipswich. |
| STACK, WILLIAM AUCHNLECK, . . . | Burton-on-Trent. |
| STOLTENHOFF, ROBERT, . . . . | Grimsby. |
| TOWNSEND, CHARLES ERIC, . . . | London. |
| WHYTE, KENNETH McINTOSH, . . . | Liverpool. |

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The PRESIDENT announced that the following eleven Transferences had been made by the Council:—

*Associate Members to Members.*

| | |
|---|---|
| BAKER, HENRY NEWTON, . . . . . | London. |
| BEALE, SAMUEL RICHARD, . . . . . | Glasgow. |
| BIGG-WITHER, LIONEL, . . . . . | Bombay. |
| BINNS, ASA, . . . . . . . . | London. |
| BRAYFIELD, THOMAS HENRY GORDON, . . . | Hong Kong. |
| HIBBERD, FREDERICK CHARLES, . . . . | Slough. |
| MACLEAN, JAMES BORROWMAN, . . . . | Madeira. |
| MARCH, SYDNEY HERBERT, . . . . . | Manchester. |

SMITH, MONTAGUE HOWARD, . . . . . London.
SPRUNT, HERBERT WILLIAM, . . . . London.
WATSON, JOHN, . . . . . . . Kilmarnock.

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The Discussion upon the two Papers on "Refrigerating Machines," which were read at the last Meeting, was resumed and concluded.

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The Meeting terminated at Ten o'clock. The attendance was 94 Members and 42 Visitors.

# THE MODULUS OF ELASTICITY, AND ITS RELATION TO OTHER PHYSICAL QUANTITIES.

By A. H. STUART, B.Sc., of London.

[*Selected for Publication.*]

It seems very improbable that such an important and well-marked physical quantity as the Modulus of Elasticity of a metal should be quite isolated and bear no relation to the other properties of that metal. The following is a record of an attempt to link up the modulus of elasticity of the metals with the other physical properties which may be reasonably supposed to be akin to it. After a very large number of trials, the modulus of elasticity was found to be a linear function of the following quantity, H,

$$\text{where } H = \frac{\text{Density} \times \text{Specific Heat}}{\text{Coefficient of Linear Expansion}}.$$

Suppose one gramme of the metal in question were taken, in the form of a rod, then H is the number of calories of heat which would be absorbed by that rod in doubling its length under the action of the heat alone.

In Table 1 (page 1156) are given the values of the quantities required for calculating H for ten common metals, and also the values of the modulus of elasticity used in the graph. These values have in every case been taken from the most reliable source, and where a quantity showed a rapid variation with a rise of temperature, its value at 50° C. was computed. (This does not apply to the values given for density.)

TABLE 1.

| Metal. | Symbol. | Density grammes per cm³. | Specific Heat. | Coefficient of Linear Expansion. | H (in calories). | E Kg. per mm² |
|---|---|---|---|---|---|---|
| Aluminium | Al | 2·58 | 0·2270 | $2 \cdot 313 \times 10^{-5}$ | $2 \cdot 54 \times 10^{4}$ | $6 \cdot 71 \times 10^{3}$ |
| Copper . | Cu | 8·9 | 0·0920 | 1·666 | 4·91 | 10·98 |
| Gold . . | Au | 19·3 | 0·0316 | 1·443 | 4·23 | 9·68 |
| Iron . . | Fe | 7·86 | 0·1130 | 1·140 | 7·80 | 18·46 |
| Lead . . | Pb | 11·34 | 0·0299 | 2·955 | 1·15 | 1·8 |
| Platinum . | Pt | 21·48 | 0·0323 | 0·899 | 7·72 | 17·04 |
| Silver . . | Ag | 10·5 | 0·0568 | 1·921 | 3·09 | 6·87 |
| Tin . . | Sn | 7·3 | 0·0559 | 2·234 | 1·83 | 4·17 |
| Zinc . . | Zn | 7·14 | 0·096 | 2·905 | 2·36 | 8·64 |
| Brass . . | — | 8·7 | 0·092 | 1·859 | 4·30 | 9·65 |

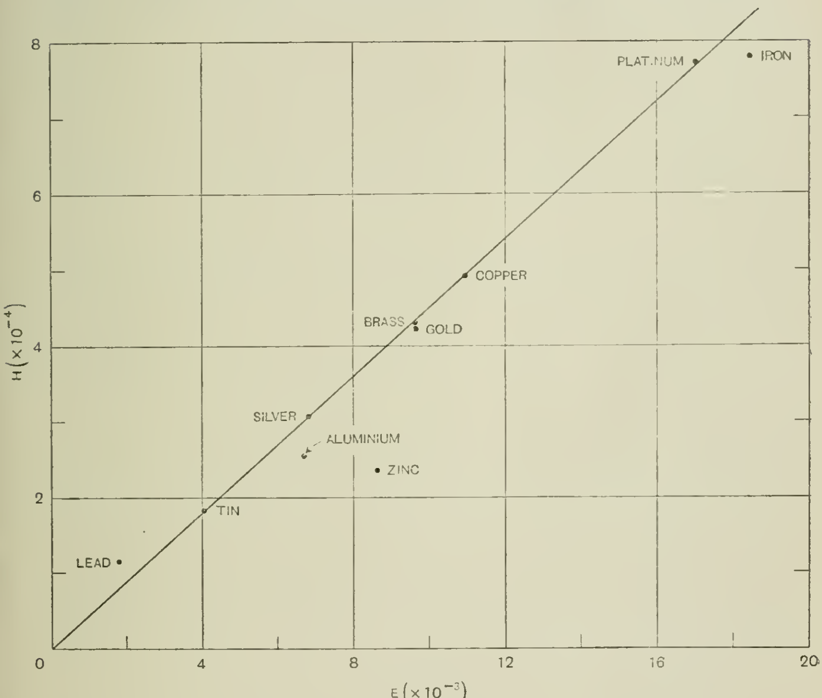
All the elementary metals of which data could be obtained have been included. Brass, the only alloy of which data referring to a sample of constant composition is at present available, is also given.

Fig. 1 shows the graph obtained by plotting H against the Modulus of Elasticity, E. It will at once be seen that, with the exception of zinc, the points lie very well about a straight line. It is significant that the line passes through the origin. This is quite in accordance with what one might expect, for if we can imagine a metal which required no mechanical energy to stretch it to double its original length, it is reasonable to suppose that no heat would be required to accomplish the same operation.

The equation for this line (determined from the graph) is $E = 0 \cdot 225\ H$. Calculating the modulus of elasticity by means of this equation, from the computed values of H, and comparing the results so obtained with those of standard observations, Table 2 is obtained (page 1158).

As already stated, zinc is the only metal which stands badly aloof from the graph. The position of lead, however, is somewhat worse than it appears, owing to its very low modulus of elasticity. The other metals which are somewhat out of alignment are aluminium and iron. Let us consider the possible explanation of these discrepancies.

Fig. 1.—*Graph obtained by plotting* H *against the Modulus of Elasticity*, E.




The modulus of elasticity of all metals varies to some extent with variations of temperature, and also with the "condition" of the specimen examined (for example, whether it is cast or drawn). The presence of impurities also influences the elasticity, especially in some metals. These variations are shown by zinc to an almost unequalled extent. The work of v. Miller shows that the modulus

TABLE 2.

| Metal. | H | E Kg. per mm² | E = 0·225 H Kg. per mm² | Error. |
|---|---|---|---|---|
| | | (Observed.) | (Calculated.) | Per cent. |
| Aluminium . . | $2{\cdot}54 \times 10^4$ | $6{\cdot}71 \times 10^3$ | $5{\cdot}73 \times 10^3$ | $+ 0{\cdot}98 = 17$ |
| Copper . . | 4·91 | 10·98 | 11·00 | $- 0{\cdot}02 = 0{\cdot}18$ |
| Gold . . . | 4·23 | 9·68 | 9·55 | $+ 0{\cdot}13 = 1{\cdot}3$ |
| Iron . . . | 7·80 | 18·46 | 17·60 | $+ 0{\cdot}86 = 4{\cdot}6$ |
| Lead . . . | 1·15 | 1·8 | 2·59 | $- 0{\cdot}79 = 43$ |
| Platinum . . | 7·72 | 17·04 | 17·30 | $- 0{\cdot}26 = 1{\cdot}3$ |
| Silver . . . | 3·09 | 6·87 | 6·96 | $- 0{\cdot}09 = 1{\cdot}3$ |
| Tin . . . | 1·83 | 4·17 | 4·13 | $+ 0{\cdot}04 = 1$ |
| Zinc . . . | 2·36 | 8·64 | 5·32 | $+ 3{\cdot}32 = 38$ |
| Brass . . . | 4·30 | 9·65 | 9·68 | $- 0{\cdot}03 = 0{\cdot}3$ |

of elasticity of zinc varies as much as 20 per cent. according to its condition, while the value measured at 0° C. falls 33 per cent. when the temperature rises to 100° C. (It will be remembered that those elements—carbon, boron, and silicon—whose specific heat varied largely with temperature, were the transgressors of Dulong and Petit's law.) Again, Zatzenelsohn has shown that the modulus of elasticity of aluminium falls 19·5 per cent. when the temperature rises from 0° to 100° C.

Lead has such a very low modulus of elasticity that any small error of observations necessarily influences the result seriously, and its ductility is such as to render its condition a very important factor.

The preparation of iron in a state of chemical purity presents considerable difficulty, and its modulus of elasticity is found to lie within very wide limits indeed, when small proportions of other substances are combined with it.

For the sake of those who wish to follow up this interesting investigation, it might be wise to mention the paths which have already been followed, but which have failed to fulfil a useful purpose. Much time was lost by taking the unit of volume as a standard of comparison; it was only when unit weight was taken that a regular graph was obtained. Again, since a rod expanded by heat increases in sectional area, while the same rod expanded under mechanical force has its sectional area diminished, it might reasonably be expected that the coefficient of bulk elasticity would be the one to use. (This coefficient is usually denoted by K and is equal to stress per unit area/strain per unit volume. It may be calculated from the formula $K = \frac{E}{3 - 6\eta}$, where $\eta$ = Poisson's coefficient.) No regular graph could be obtained by its use, however.

The Paper is illustrated by one Fig. in the letterpress.

## MEMOIRS.

EDWIN ADAMS was born in Manchester in 1862. He received his technical education at the Owens College, Manchester, where he was twice placed first in the examination for the Ashbury Exhibition. He served his apprenticeship from 1880 to 1887 in the locomotive workshops of the Manchester, Sheffield, and Lincolnshire Railway at Gorton, and on its completion he was engaged as a draughtsman in Messrs. Hulse and Co.'s Works, Salford. During 1888–9 he took charge of the firm's exhibits at Melbourne, and on his return to Salford he was employed as foreman at the works until 1892, when he was appointed assistant manager. In 1894 he became outside representative of the firm, both at home and abroad, and on its conversion into a company in 1898 he was made a director. This position he held until the death of Mr. Henry Bates in 1903, when he was appointed managing director; he retained this post until his death which took place at Alderley Edge on 28th September 1912, at the age of fifty. He became a Member of this Institution in 1897.

ASPLAN BELDAM was born at Bluntisham, Hunts, on 5th October 1841. He served his apprenticeship with Messrs. Kitson, Thompson and Hewitson, locomotive engineers, of Leeds, since which time his life was spent wholly in marine engineering work. He gained his first few years' experience at Messrs. Miller and Ravenhill's Works, Blackwall, Messrs. Lungley's, Deptford, with the General Steam Navigation Co., and with Messrs. John Penn and Sons, Greenwich, a portion of this time being spent in the engine-room at sea. In 1865 he left the service of Messrs. Penn and Sons, and took the position of manager of the City of Worcester Locomotive and General Engineering Works. Two years later he was appointed manager of shipbuilding and engineering works at Northfleet,

where he built and engined two merchant steamers on the compound principle; and in 1871 he brought out a boiler to work at a pressure of 150 lb. per square inch. At the end of 1869 he joined the firm of Messrs. George Forrester and Co., Vauxhall Foundry, where he carried out some important contracts. A few years later he accepted the position of superintendent engineer of the Flower Line of steamships; and in 1876 he commenced practice in London as consulting engineer. In this capacity he acted for the Castle Line (Messrs. Thomas Skinner and Sons), Messrs. Money Wigram and Sons, the Eastern Telegraph Co., and many other firms. Amongst the well-known steamers built to his designs and under his supervision was the S.S. "Stirling Castle," which brought home from Hankow upwards of 6,000 tons of tea in the unprecedented time of twenty-eight days. He was the founder of the business of the Beldam Packing and Rubber Co., and brought out many useful inventions, such as semi-metallic packings, metallic rings for high-pressures, and corrugated metallic valves for air and circulating pumps. His death took place at his residence at Ealing on 16th December 1912, at the age of seventy-one. He became a Member of this Institution in 1888; he was also a Member of the Institution of Naval Architects, and was the first President of the Institute of Marine Engineers.

Joseph Ashworth Boorman was born at Heywood, Lancashire, on 3rd December 1855. He was educated at the Lancastrian School and Mechanics' Institute, Manchester, and the Manchester School of Art evening classes. At the age of fifteen he began an apprenticeship of seven years at the works of Messrs. J. and J. Kershaw and completed it at the machine-tool works of Messrs. Craven Brothers, Manchester. In 1877 he was engaged as assistant foreman with Messrs. John Elce and Co., machinists, Manchester, and two years later he became chief foreman in the tool department of Messrs. Greenwood and Batley, Leeds. This position he held until 1888, when he was appointed chief representative for the firm. After having been over thirty-one years with Messrs. Greenwood and Batley, he resigned his post in 1910 to commence

business on his own account in Leeds as engineering merchant and agent; and was making good progress in building up a connection when his death took place suddenly, at Blackpool, on 31st August 1912, in his fifty-seventh year. He became a Member of this Institution in 1895.

Henry Wheeler Bulkley was born in New York City on 22nd July 1841. He received his preliminary education in the public schools of New York, and entered the College of the City of New York. Shortly after the beginning of the Civil War in 1861, he left college to receive a commission in the Navy as assistant engineer. He served throughout the remainder of the war, and, at its termination, opened offices as an engineer in New York City in 1866. Shortly after this, he perfected his invention of an injector-condenser, and from that time until his death he was engaged in its manufacture, as well as that of steam-pumps, pyrometers, and water-heaters. He also made numerous improvements in connection with steam appliances, pumps, &c. His death took place on 8th November 1911, at the age of seventy. He became a Member of this Institution in 1881.

Dugald Drummond was born at Ardrossan on 1st January 1840, his father being permanent-way inspector on the Bowling section of the North British Railway. He served his apprenticeship with Messrs. Forrest and Barr, general engineers and millwrights, of Glasgow, and on its completion he worked on the Caledonian and Dumbartonshire Railway for a time, and then spent two years with Mr. Thomas Brassey, of Canada Works, Birkenhead. In 1864 he joined the Edinburgh and Glasgow Railway Co., prior to its amalgamation with the North British Railway Co., and in the following year he went to Inverness as foreman erector in the Highland Railway Works, under the late Mr. William Stroudley. In 1870 Mr. Stroudley left the North in order to take up the appointment of locomotive, carriage and wagon superintendent of the London, Brighton and South Coast Railway, and Mr. Drummond resigned his post at Inverness in order to accompany

him as his assistant. Five years later he was offered the post of locomotive, carriage and wagon superintendent of the North British Railway, which he accepted. In 1882 the Caledonian Railway Co. asked him to fill the same position on its line, and he accordingly left the service of the North British Railway Co. to take up his new post, which he held for eight years. In 1890 he finally severed his direct connection with the Scottish railway companies, and set up in business on his own account, founding the Glasgow Railway Engineering Co., at Govan, which is still being carried on by his surviving son, Mr. George Drummond. In 1895 he was offered and accepted the appointment of chief mechanical engineer of the London and South Western Railway Co., which had previously been held for seventeen years by Mr. W. Adams. This appointment he held—during the same period as his predecessor—until the time of his death. Among the best known of his many inventions may be mentioned the cross-tube locomotive fire-box, its average life being not less than equivalent to a locomotive mileage of 350,000; boilers fitted with these fire-boxes are notable for their easy steaming. Another feature he introduced with success was the heating of the water in the tender by means of exhaust steam. Such high temperatures as were attained rendered the use of injectors impossible, and accordingly he installed steam-operated feed-pumps. All the engines on the London and South Western Railway are now fitted with the Drummond spark arrester, which is one of the most efficacious of any that have been tried in this country. One or two of the different types of locomotives he introduced may be referred to. In 1898 he designed a locomotive (No. 720) purposely to do away with piloting. It really consisted of two entirely independent sets of engines driving separate shafts, the wheels themselves being without coupling-rods. Another engine which embodied several new departures was the six-coupled four-cylinder non-compound locomotive which was introduced at the end of 1907. This engine with its tender weighed as much as 116 tons; it contained a steam-drier in the smoke-box, and the tender was fitted with a water picking-up arrangement. In 1903 he introduced a steam motor-coach to deal

with the midday suburban traffic. The first to be made was intended to run between Fratton and Havant. The removal of the locomotive works from Nine Elms to Eastleigh was carried out under his supervision, and the new works were erected to his designs. He was a man of great ability and gifted as an organizer and manager of men. During the late railway strike not a single man from his department went on strike. He established classes in the works for the training of the young men under him, and took a keen interest in their welfare. His death took place at his residence in Surbiton, after a serious operation following a scald on the leg, on 7th November 1912, in his seventy-third year. He became a Member of this Institution in 1886; he was also a Member of the Institution of Civil Engineers, and of the Association of Railway Locomotive Engineers, and held the rank of Major in the Engineer and Railway Staff Corps, Territorial Force.

Finlay Finlayson was born at Dunbar, East Lothian, on 10th April 1852. He was educated in the same town, and at the age of eighteen he went to the west of Scotland to commence an apprenticeship with Messrs. Miller and Anderson, Atlas Works, Coatbridge. On its completion in 1874 he remained with the firm, being appointed shop foreman, a position he held until he received an appointment in 1881 as chief of the erecting department in the works of Messrs. Easton and Anderson, Erith. Later in the same year he went to the Mossbay Iron and Steel Works, Workington, to take up the position of works engineer. Two years later he was selected by the Glengarnock Iron and Steel Co., to superintend and lay out the new steel works which they were erecting at that time. On the completion of this undertaking, he returned to his old firm in Coatbridge—now known as Miller and Co., Vulcan Foundry—to be general manager. After being with them for ten years, he was appointed by Messrs. Stewarts and Menzies to take charge of the remodelling of their Clydesdale Steel Works, where he remained for five years. In 1902 he commenced business on his own account as a consulting engineer in the Iron Works District, and carried

out some important new works, including Cairnhill Iron Works, Coatbridge, and a large engineering shop and Siemens furnace for Messrs. R. B. Tennent, Ltd., of Coatbridge. His death took place, after a long illness, on 12th October 1912, at the age of sixty. He became a Member of this Institution in 1891; he was also a Member of the Iron and Steel Institute and the Institution of Engineers and Shipbuilders in Scotland.

GEORGE HYDE was born in Manchester on 18th November 1868. He was educated at Derby, and was engaged in the Derby and Nottingham District Engineer's office, Great Northern Railway, under Mr. A. J. Grinling, as assistant upon the varied works in connection with 150 miles of railway and 50 miles of canal. This work comprised the preparation of drawings, surveys, and setting out of branch railways and extensions, and supervising their construction, including that of bridges and signalling work. In March 1898 he was appointed chief engineer and works manager to Messrs. Pilkington Bros. Sheet Glass Works, St. Helens, Lancashire. In this capacity he had charge of the construction of large Siemens gas-furnaces, gas-producers and plant, boilers, clay-working machinery, Dellwik and Mond plants, &c. This position he held until his death which took place at St. Helens, after a prolonged illness, on 14th November 1912, in his forty-fourth year. He became an Associate Member of this Institution in 1902, and was an Associate Member of the Institution of Civil Engineers; he was also a Member of the Council of the Liverpool Engineering Society.

ROBERT MIDDLETON was born at Leeds on 4th October 1847. At the age of twenty-three he became the proprietor of the Sheepscar Foundry, Leeds, making a speciality of hydraulic machinery. The business rapidly increased, so that the works had to be enlarged, and the building of Corliss, pumping, and stationary engines was added. Subsequently he brought out machinery for making patent fuel or briquettes, and developed a large business in this connection. His death took place at Leeds

on 11th October 1912, at the age of sixty-five. He became a Member of this Institution in 1891.

Vaughan Pendred was born in Ireland in 1836, and lived until a young man on the family estate of Barraderry in County Wicklow. His father was a classical scholar of some note and his mother a highly educated woman. From them and from two resident governesses he received all his teaching. He was never at a school, and his education was due to his extraordinary avidity for books, aided by a highly retentive memory. The estate, being Irish, was impoverished and he had to make his way in the world alone, without financial assistance of any kind and without any greater preparation for an engineer's calling than he had acquired by reading, study, and the making of models. He was known to Zerah Colburn, the editor of *The Engineer* at that time, by his letters, and when Mr. Pendred came to London to look for work as an engineer, Colburn introduced him to Mr. Aveling, of Aveling and Porter's, and Mr. Aveling found him an opening in a little works in Staffordshire. But his letters on engineering subjects had attracted the attention of Mr. Passmore Edwards, the proprietor of *The Mechanic's Magazine*, and, after a few months in the works, he was asked to fill the editorial chair of that paper. Two years or so later, in 1865, when Mr. Colburn left, Mr. Pendred was offered the editorship of *The Engineer*, and retained it till 1905 when he retired, his second son—Loughnan Pendred, *Member*—taking his place. The history of his life's work is to be found in the pages of the paper to which he devoted all his energies. He knew all the eminent engineers of his time and had seen the whole development of engineering which marked the Victorian era. Traction by sea and land appealed to him particularly, and his opinion on railway matters and the locomotive especially, commanded universal respect. He was a welcome visitor at the meetings of all engineering societies, and not infrequently took part in discussions which he illuminated by his sound common sense and brightened by the humour which never failed him. His death took place at his residence at Streatham, where he had

lived for forty years, on 12th October 1912, at the age of seventy-six. He became a Member of this Institution in 1900; he was also a Member of the Iron and Steel Institute, and for many years of the Society of Engineers, and was an honorary member of the Cleveland Institution of Engineers.

CHARLES DAVID PHILLIPS was born at Newport, Mon., on 26th December 1845. After having been educated at Long Ashton and Normal College, he served his apprenticeship as a mechanical engineer, and made a rapid advancement so that he soon became proprietor of an engineering establishment. He ultimately took extensive premises at Newport, known as the Emlyn and Central Engineering Works, together with works at Gloucester, and subsequently added branches in London and Cardiff. At the Newport Works he carried out repairs of all kinds to locomotives, engines, boilers, &c., and manufactured hauling engines, saw-benches, mortar mills, foundry core ovens, &c. He also brought out apparatus for automatically controlling Bessemer converters. His entire Engineering Works and Foundry have recently been concentrated at Newport, Mon. All phases of outdoor life appealed to him. He not only manufactured and dealt in agricultural machinery, but became an active farmer in a large and successful way; and for a number of years he was one of the honorary secretaries of the Monmouthshire Chamber of Agriculture. In addition to his connection with local societies, he was a very active participant in the Shows of the Royal Agricultural Society and the Bath and West of England Society. For a few years he was a member of the Newport Corporation, but retired through pressure of business, though he continued to take a deep interest in the public affairs of the town and county, for both of which he was a Justice of the Peace. His death took place at his residence in Newport, after a long illness, on 21st October 1912, in his sixty-seventh year. He became a Member of this Institution in 1885; he was also a Member of the Iron and Steel Institute and of the South Wales Institute of Engineers.

Walter Henry Scott was born at Coldstream, Berwick, on 16th January 1837. His early education was received at Kincardine-on-Forth, after which he studied at the City of London School. In June 1851 he began his apprenticeship under the late Mr. J. E. McConnell at the Wolverton Engine Works of the Southern Division of the London and North Western Railway. After passing through the various departments, including the drawing office, he was transferred to the office of the locomotive superintendent, and subsequently was promoted to the post of outdoor inspector in the locomotive department. When the Southern Division was amalgamated in 1862 with the Northern Division, he was transferred to Crewe. In December 1863 he was recommended by Mr. John Ramsbottom to Sir John Hawkshaw for the post of locomotive and carriage superintendent of the Mauritius Government Railways, which post he held from January 1864 to June 1870 when he was appointed General Manager and Engineer of these railways. During the period he held this position the two spans of the Grand River Bridge, damaged by a hurricane in March 1868, were repaired, launched, and fixed, and various extensions to the lines were made. For these services he received the special thanks of the Executive Council. In 1879 he resigned his appointment, and after a few months as assistant manager of the Mersey Steel and Iron Works, Liverpool, he proceeded to Chile as general manager and engineer of the Taltal Railway. This was a line abounding in steep gradients and sharp curves and with almost a complete absence of water suitable for locomotives. After about three years the concern was rendered successful, and Mr. Scott went to Buenos Aires as locomotive, carriage and wagon superintendent to the Buenos Aires Great Southern Railway. Six years later he was appointed general manager and engineer of the Northern of Uruguay Railway (now amalgamated with the Midland and North Western of Uruguay Railway). Having returned to England, he was offered the post of general manager and engineer of the Great Western of Brazil Railway, but not agreeing with the policy of the directors he retired. Since that date, 1904, he resided at

East Molesey, Surrey. His death took place at Ilford, on 8th November 1912, in his seventy-sixth year. He became a Member of this Institution in 1861.

Walter Henry Smith was born in Manchester on 2nd June 1857. His scholastic education was received at St. Saviour's elementary school, Manchester, and his technical education was self-taught. In 1867 he began an apprenticeship of seven years in the shops and drawing office of Messrs. Sharp, Stewart and Co., Atlas Works, Manchester, and on its completion in 1874 he was engaged as draughtsman at the works of Messrs. Beyer, Peacock and Co., of Manchester. Six years later he became head draughtsman with Messrs. Kitson and Co., Airedale Foundry, Leeds, and in 1884 he was appointed chief draughtsman and assistant manager with Messrs. Hawthorn, Leslie and Co., Newcastle-on-Tyne. From 1890 to 1897 he acted as inspector for the late Sir James Brunlees, and was also in business on his own account as a consulting engineer. In the latter year he was appointed works manager with Messrs. Kerr, Stuart and Co., Stoke-on-Trent, which position he held until 1899, when he acted for two years as consulting engineer at Leeds and Blackpool. In 1901 he went to South Africa to take up the position of mechanical engineer to the Public Works Department of the Government of Cape Colony, from which position he retired in 1908. His death took place on 29th March 1912, in his fifty-fifth year. He became a Member of this Institution in 1904.

William Tartt was born at Trowbridge on 29th February 1832. He served his apprenticeship at the Coalbrookdale Iron Co.'s Works, and on its completion he was employed for a time in the locomotive department of the London and North Western Railway at Wolverton and Crewe. He next was engaged in the works of Messrs. J. and G. Rennie, marine engineers, of London, until 1864 when he was appointed superintending engineer of the Euphrates and Tigris Steam Navigation Co. In that capacity he had complete charge of all the company's steamers and engineering works on those two

rivers, his headquarters being at Bussora and Bagdad. In 1883 he resigned his position and returned to this country, taking up his residence at South Godstone, Surrey. He took a great interest in local affairs, and for 25 years was a member of the Godstone Rural District Council and Board of Guardians, of which latter body he was chairman for some years. His death took place suddenly at his residence, on 9th August 1912, in his eighty-first year. He became a Member of this Institution in 1879.

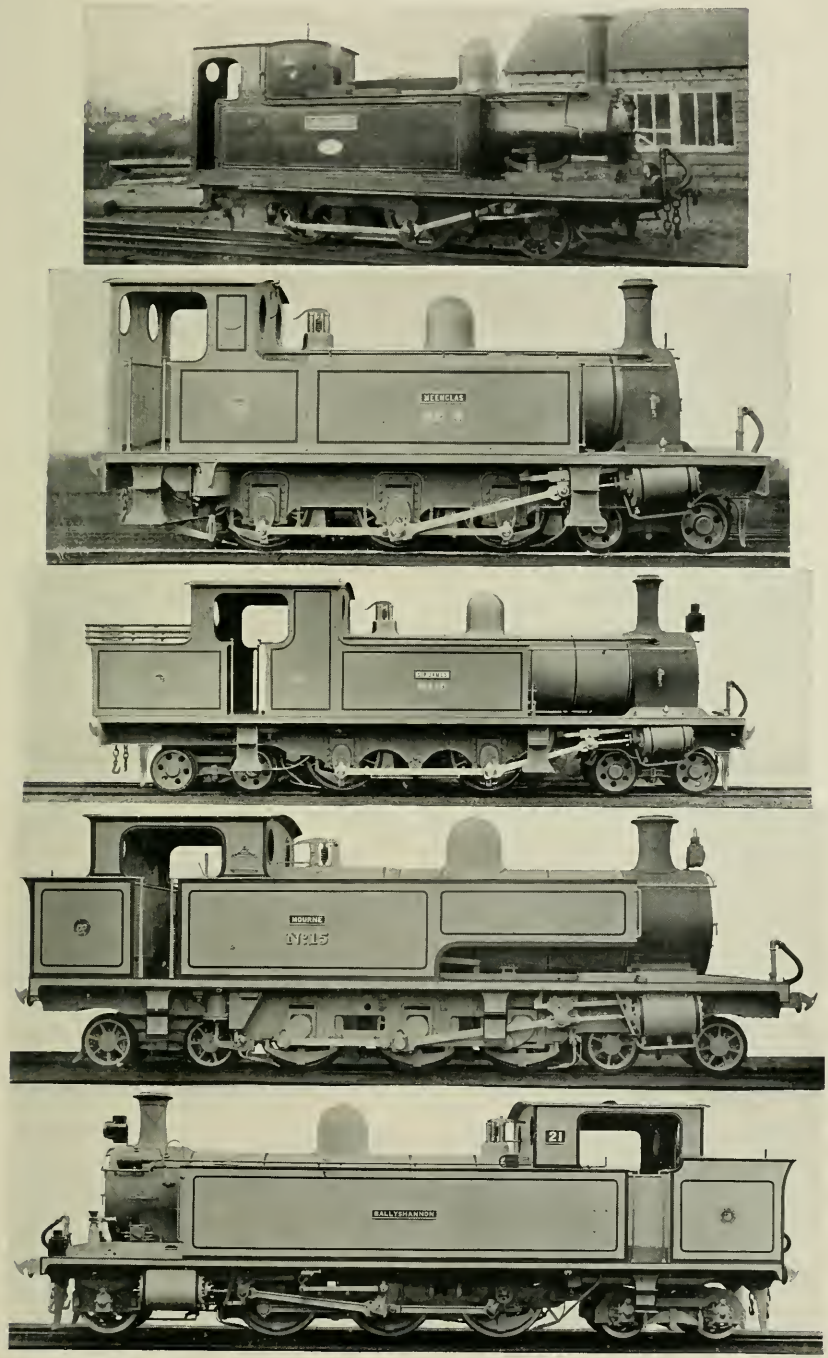
WILLIAM JOHN WAKEFIELD TURVEY was born at Plumstead, Kent, on 25th November 1881. After being educated at private schools, he commenced an apprenticeship at the age of fifteen, in the Torpedo Factory, Royal Arsenal, Woolwich, and in the following year he was transferred to the Torpedo Design Office, which position was obtained by competitive examination. During the six years he was in this department he also studied in the evenings at the Woolwich Polytechnic, where he obtained several diplomas and prizes. In October 1903 he became draughtsman at the India Rubber, Gutta Percha and Telegraph Works Co., Silvertown, and in October of the following year he went for a short time as draughtsman at Messrs. Robert Hoe and Co.'s Works, Borough Road, London. In February 1905 he returned to the Royal Arsenal as draughtsman in the Chief Mechanical Engineer's Department. Four years later he resigned his position and went to New Zealand, residing at Napier, when he became assistant to the borough engineer in March 1911. In October of the same year he was appointed assistant town clerk, and held that position until his death which took place after a short illness, on 28th August 1912, in his thirty-first year. He became an Associate Member of this Institution in 1910.

# INDEX.

1912.

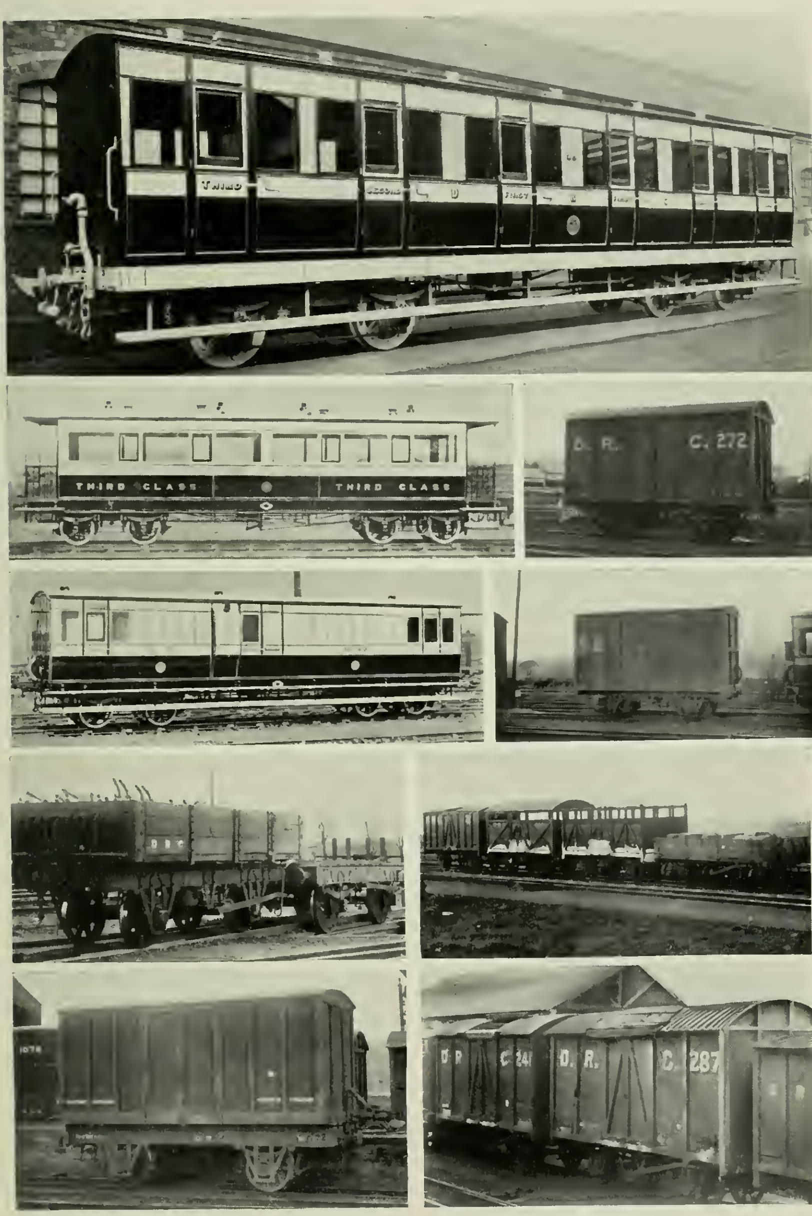
Parts 3–4.

Fig. 13. *County Donegal Railways.*
*For Dimensions of all Locomotives see Table 1 (page 624).*

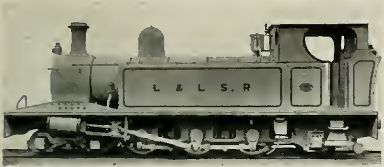


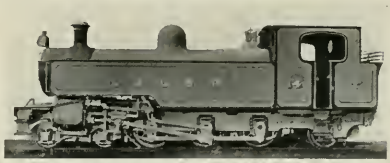
*County Donegal Railways.*

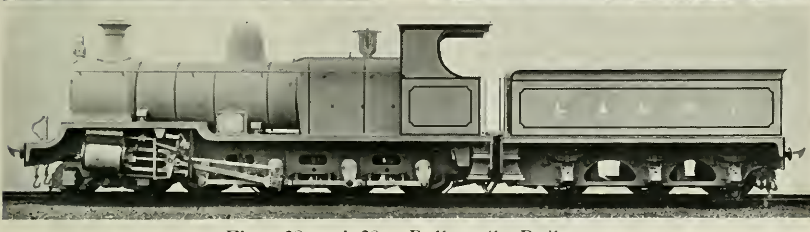
Fig. 14. *Bogie Passenger Stock, Tranship Truck on narrow gauge Underframe, also Goods, Cattle and Horse Boxes.*



Figs. 21, 22 and 23. *Londonderry and Lough Swilly Railway.*





Figs. 28 and 29. *Ballycastle Railway.*

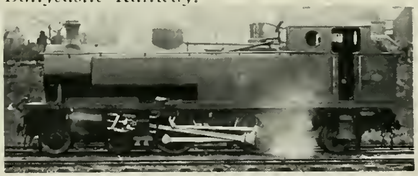


Fig. 31. *Cork, Blackrock and Passage Ry.*

Fig. 34. *West Clare Railway.*

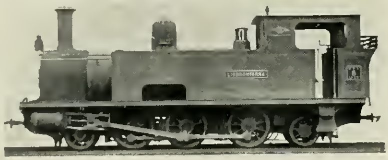
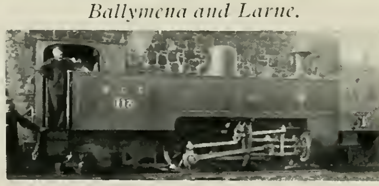


Fig. 35. *West Clare Railway.*

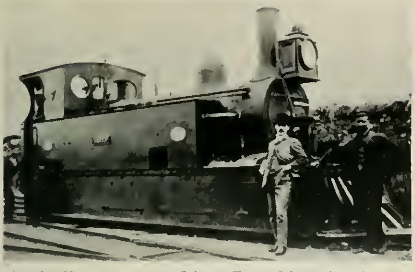
Fig. 36. *Schull and Skibbereen Ry.*

Fig. 40. *Midland Ry. (N.C.C.) Ballymena and Larne.*

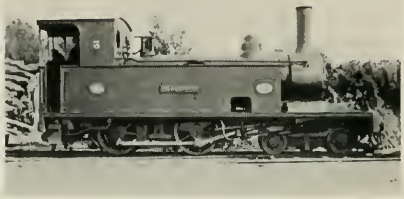


Figs. 45 to 48. *Cavan and Leitrim Railway.*

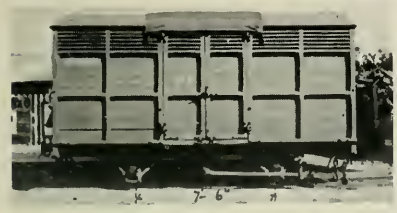
*Tramway Engine fitted to be driven from either end.*



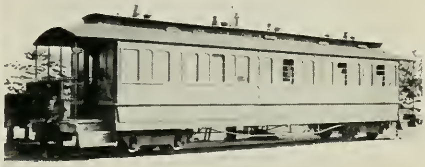
*4—4—0 Engine.*



*Goods Wagon to carry 5 tons, Tare, 3 tons, 1 c., 1 q.*



*3rd class Carriage, 50 Passengers, Tare, 7 tons, 12 c.*



Figs. 54 and 55. *Castlederg and Victoria Bridge Tramway.*

*Passenger Coach, Length 19 ft. 9 in. Width 6 ft. Height from rail 9 ft. 6 in. Wheel-base, 7 ft. 6 in.*


*Covered Goods Wagon, Length 13 ft. 6 in. Width, 6 ft. 6 in. Height from rail, 9 ft. 6 in. To carry 5 tons. Tare, 4 tons.*

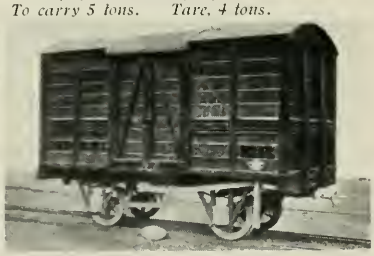


Fig. 56. *Listowel and Ballybunion Railway.*

Fig. 62. *3 ft. 6 in. Gauge Tank Locomotive.*
*(Mr. H. W. Garratt's communication.)*

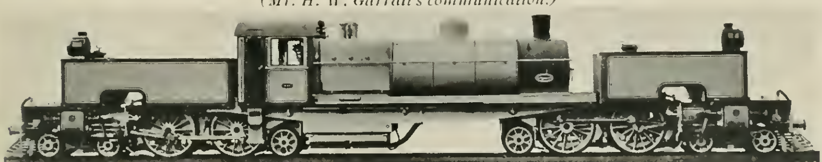


Plate 27.

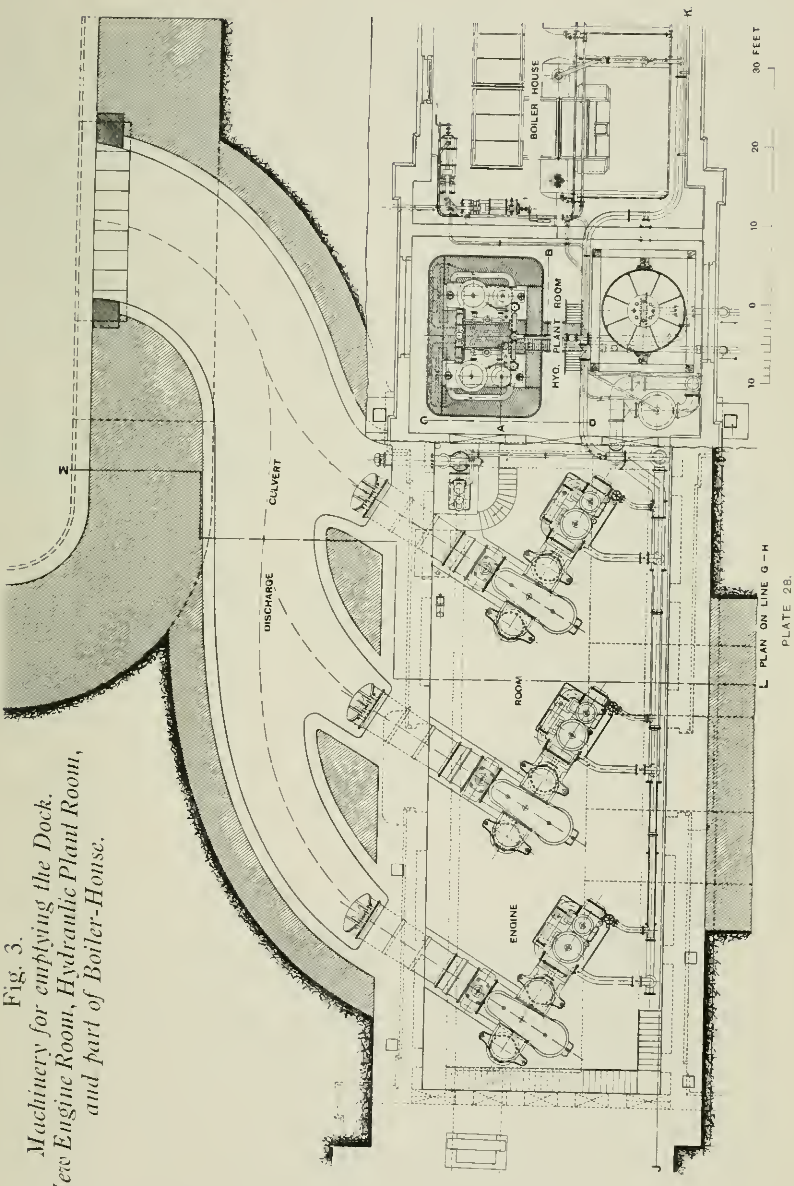
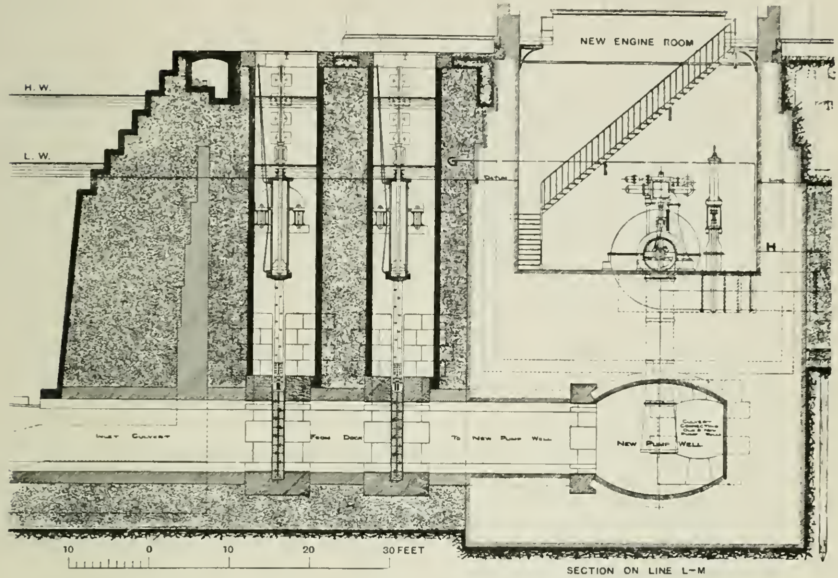
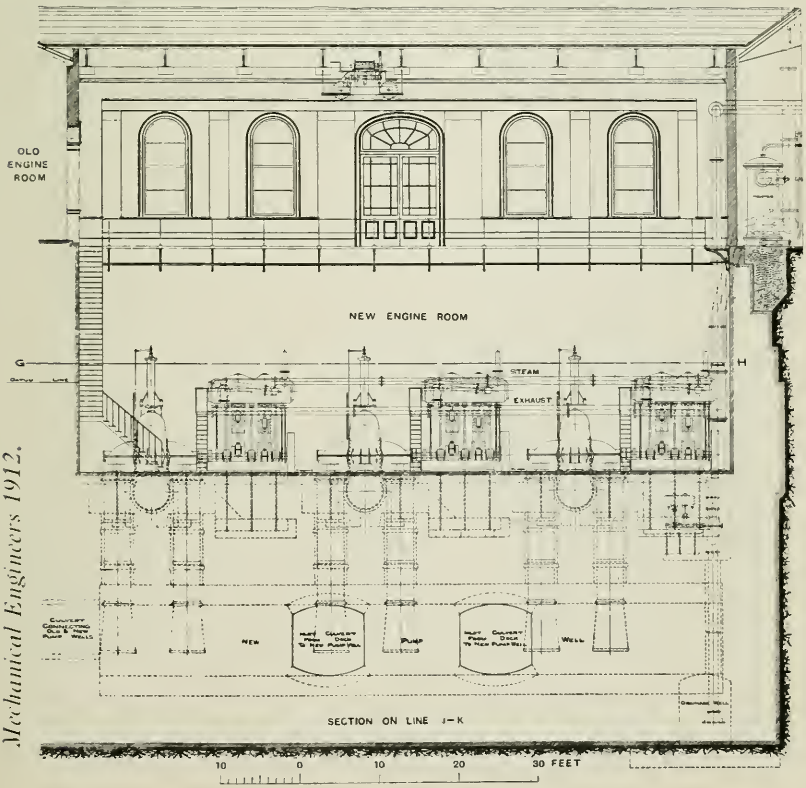



Fig. 3.
*Machinery for emptying the Dock.*
*New Engine Room, Hydraulic Plant Room,*
*and part of Boiler-House.*

Figs. 4 and 5. *Machinery for emptying the Dock.*
*New Engine Room.*

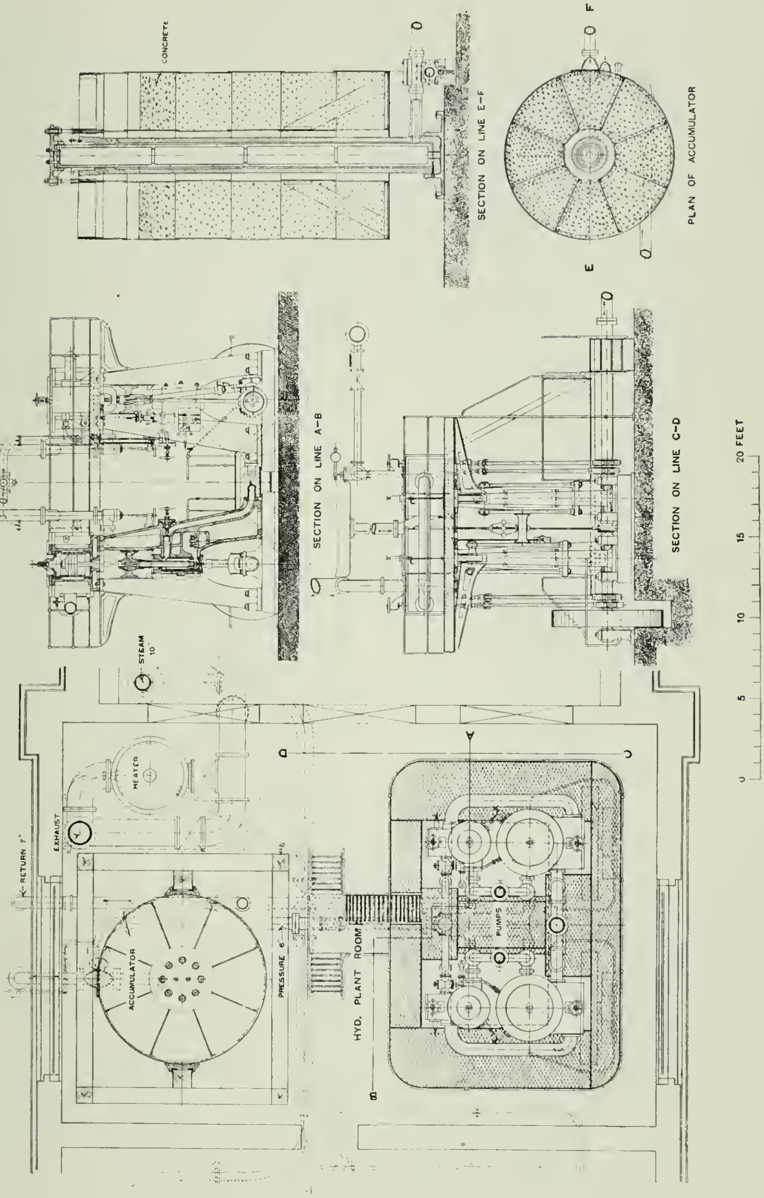



SECTION ON LINE L–M
PLATE 27.




SECTION ON LINE J–K

Fig. 6. Hydraulic Power-Supply Plant.



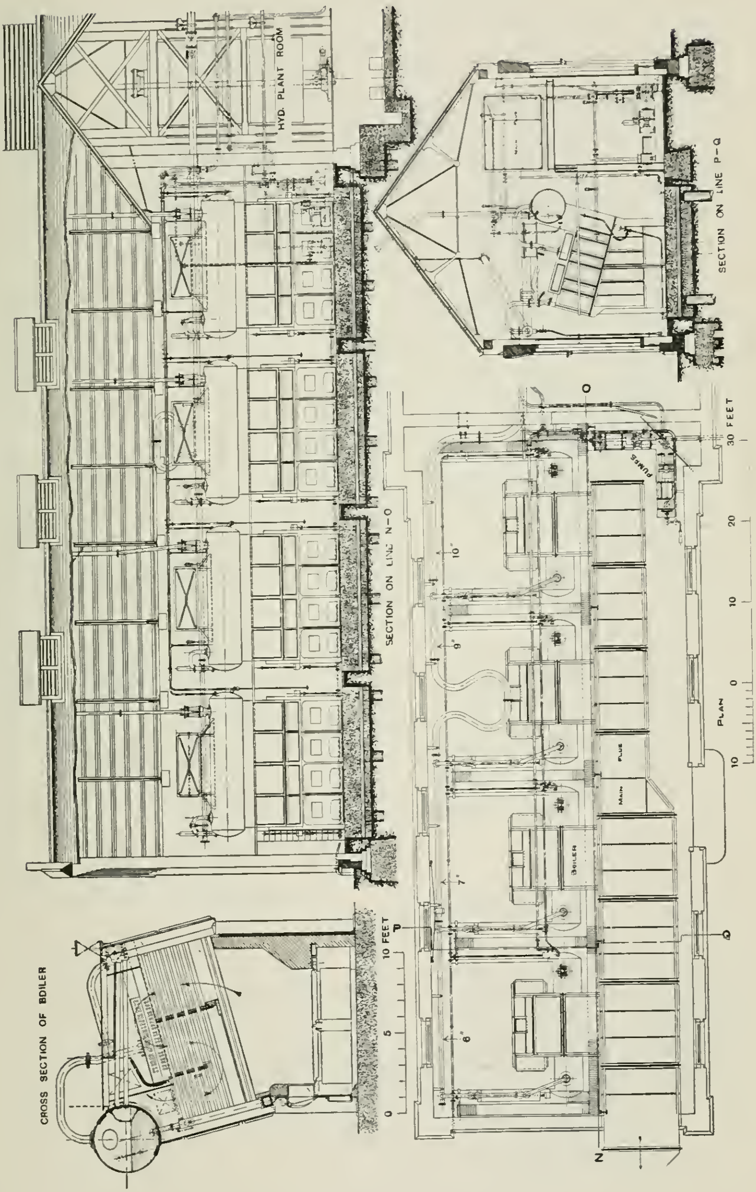
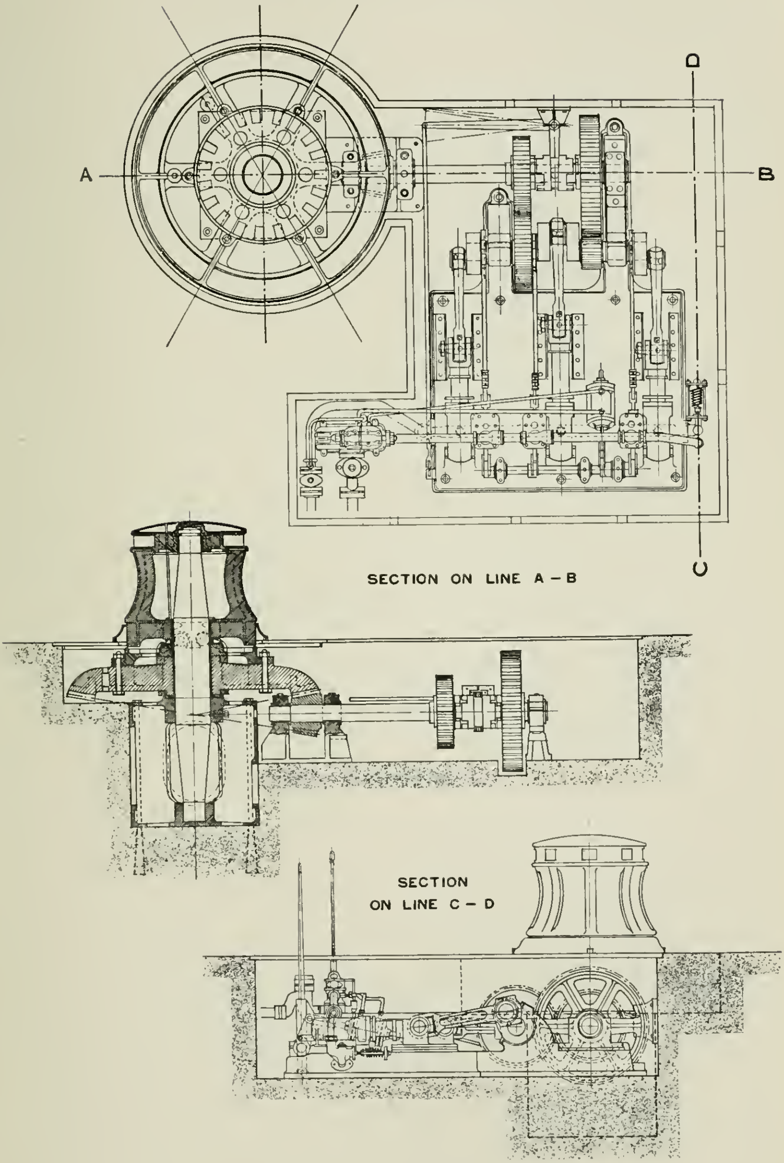



Fig. 7. Boilers, Feed-Heater, and Steam-Piping.

Fig. 8.

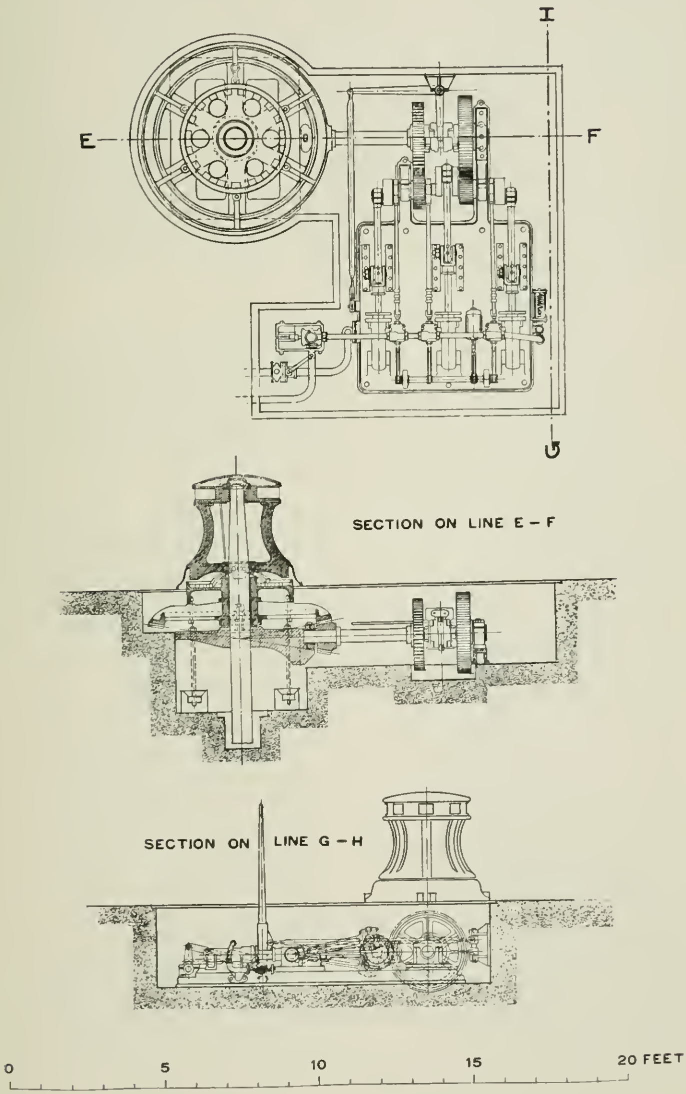
*30-ton Hydraulic Capstans.*




*For Scale, see Plate 32.*

Fig. 9.

*11-ton Hydraulic Capstans.*



# BELFAST NEW GRAVING DOCK.

Plate 33.

Fig. 10. Caisson.

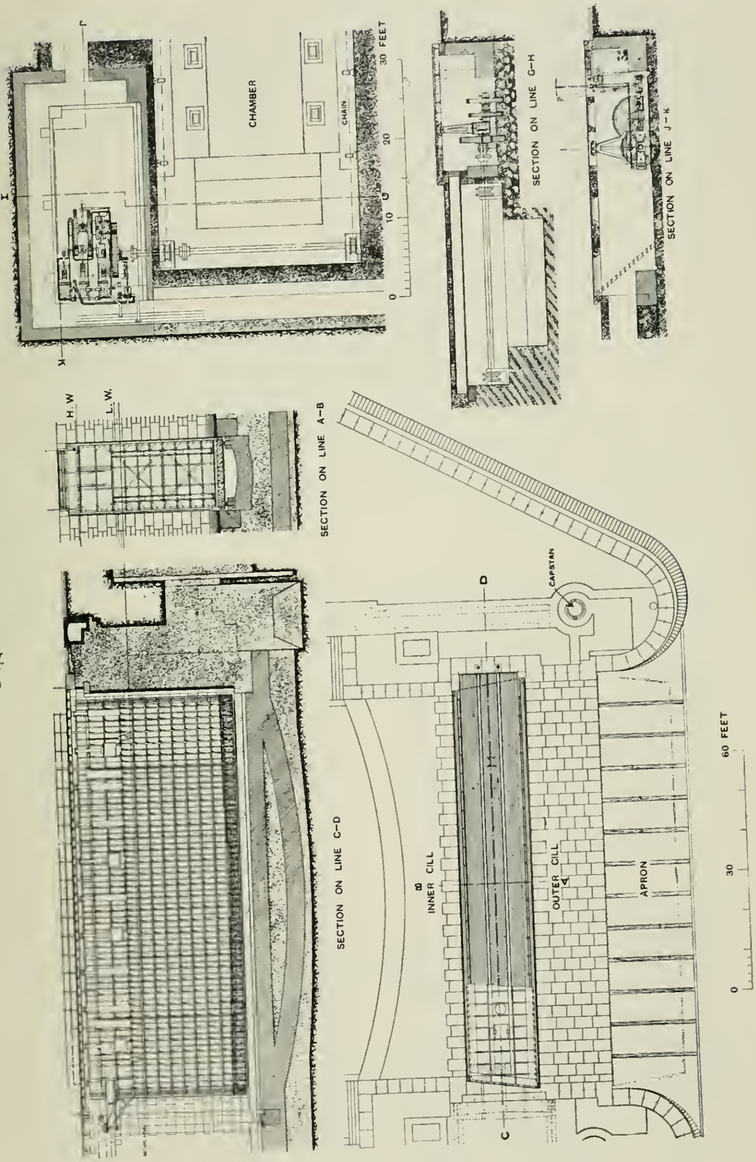


Fig. 11. Penstock Doors for Culverts.

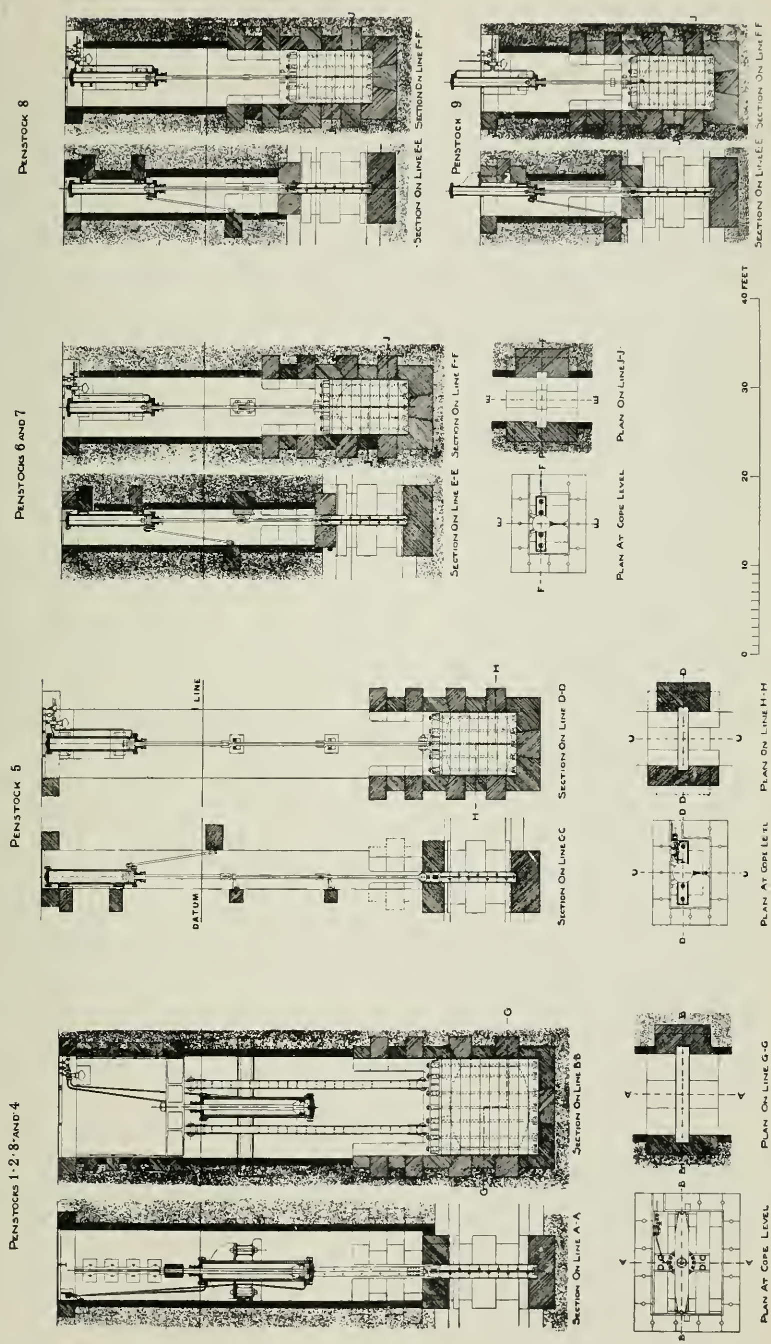


Fig. 1. *Wooden Spindles and Whorls.*


*a.* *Peru.*
*b.* *Egypt.*
*c.* *S. Italy.*
*d e f.* *Russia.*
*g.* *Germany.*
*h.* *Congo.*
*i.* *Nigeria.*
*k.* *Madeira.*

*Scale for all except e and g.*

Fig. 2. *Wooden Spindle, and Whorl made from a coin, India.*


*Spinning Wheels.*

Fig. 3. *India.*

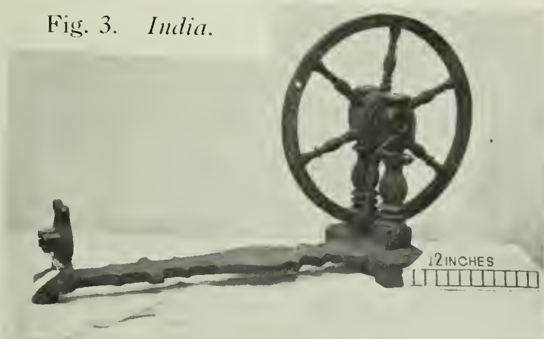
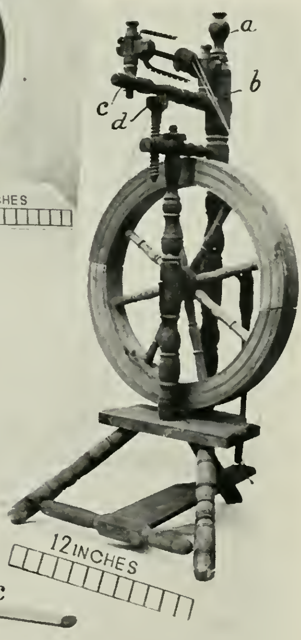



Fig. 5.



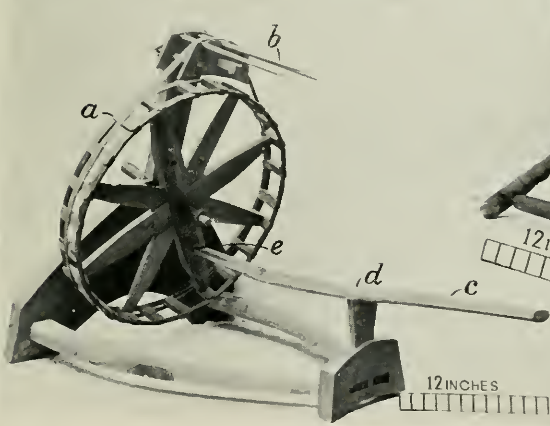



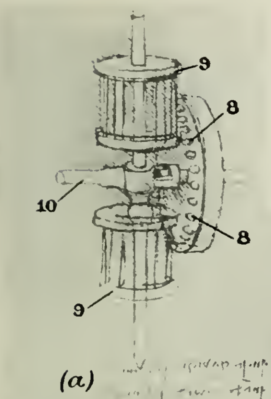

Fig. 4. *China.*

# FLAX SPINNING SPINDLE.



*Sketch of a Spinning Wheel by Leonardo da Vinci, died 1519.
Presented to the Ambrosian Library in Milan, 1639.*

Fig. 8.



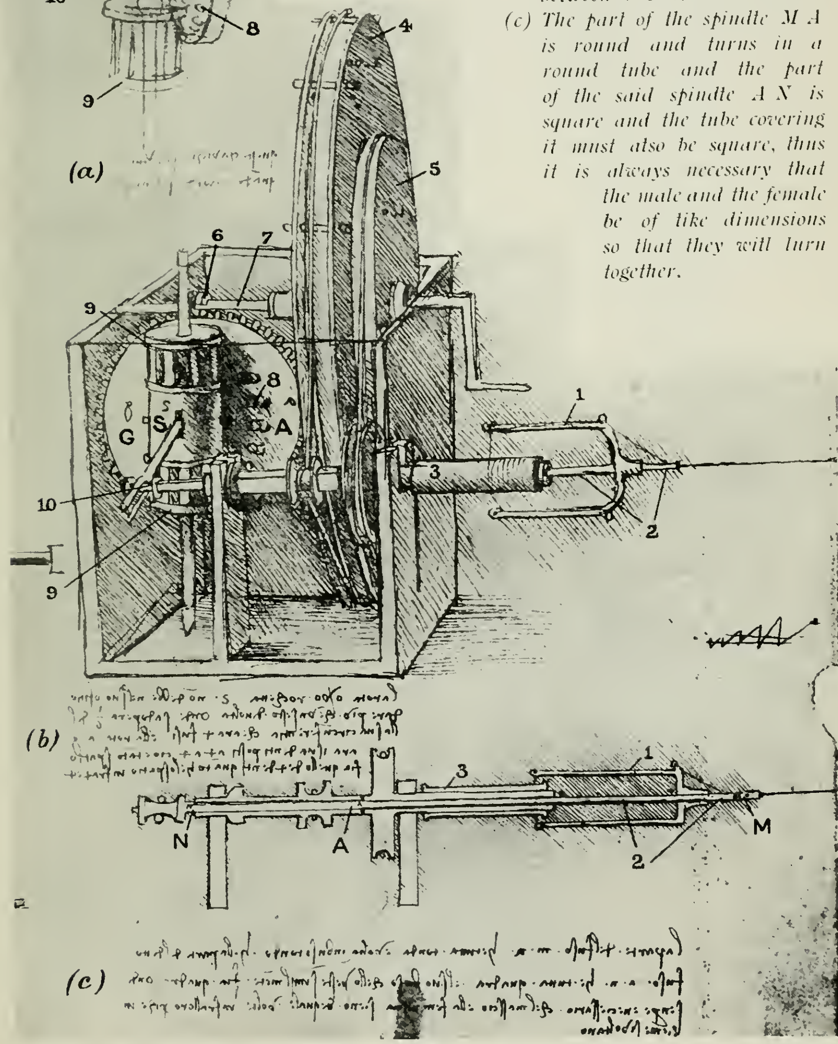

*Translation.*

*(a) The pins (Fig. 7) are put in for tightening the band.*

*(b) The wheel or ratchet S must not give more than one sixth of a turn, thus using one sixth of its circumference which has 4 cogs; the wheel A G must have the teeth placed 4 × 4 that is as much space between the wheel for 4 teeth as in the space between 4 & 4.*

*(c) The part of the spindle M A is round and turns in a round tube and the part of the said spindle A N is square and the tube covering it must also be square, thus it is always necessary that the male and the female be of like dimensions so that they will turn together.*

Fig. 7.



## FLAX SPINNING SPINDLE.

*Spindle and Fliers attached to Bearings.*

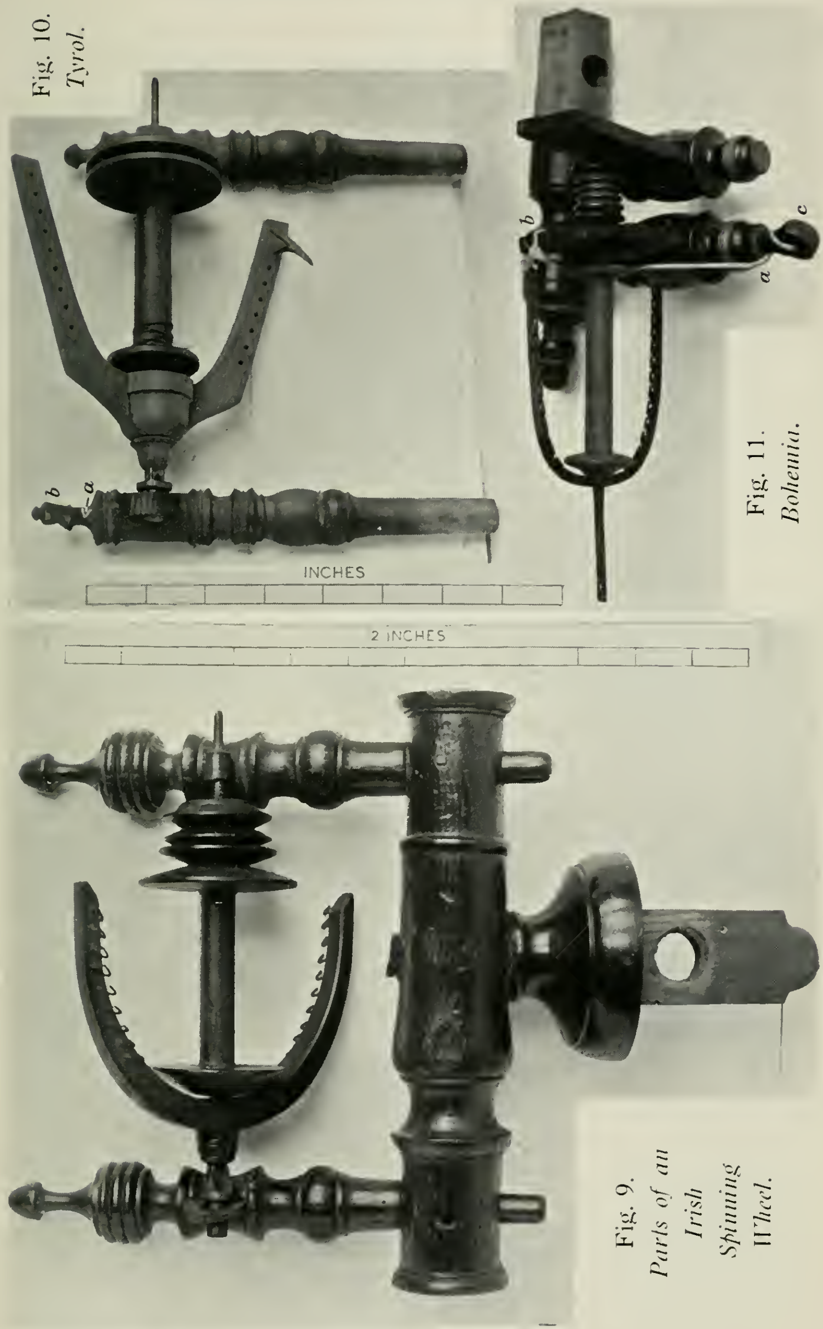


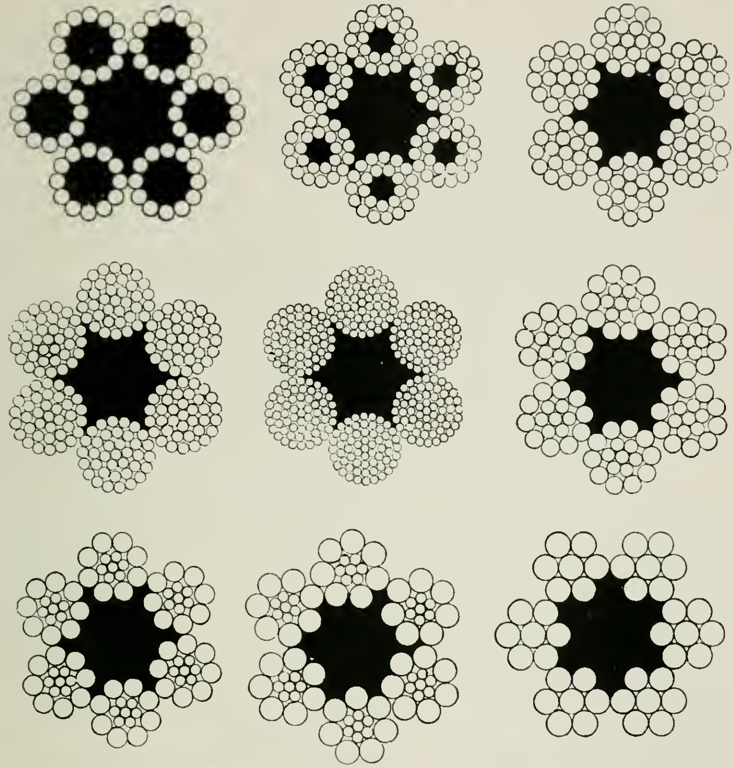

Fig. 10. *Tyrol.*

Fig. 11. *Bohemia.*

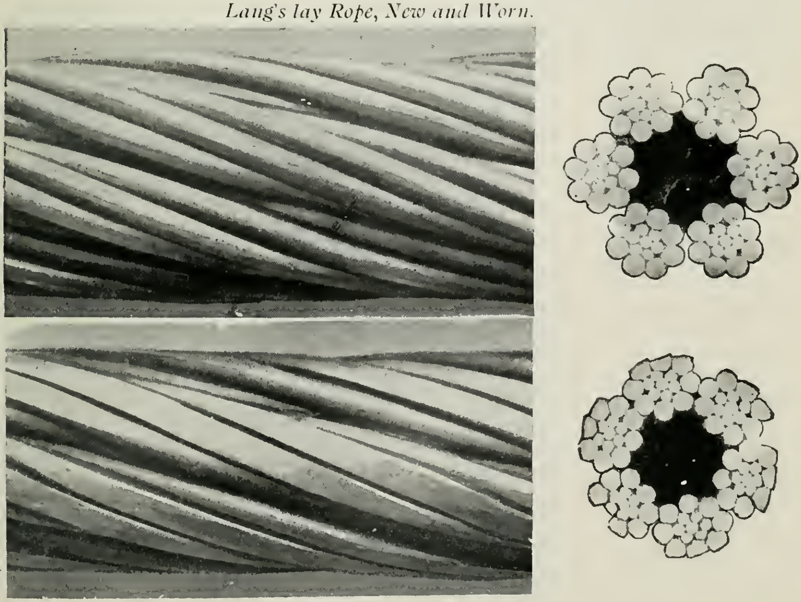
Fig. 9. *Parts of an Irish Spinning Wheel.*

*(Mr. H. Lowthian Barge's communication.)*

Fig. 15. *Nine Ropes showing various numbers of wires in each strand.*

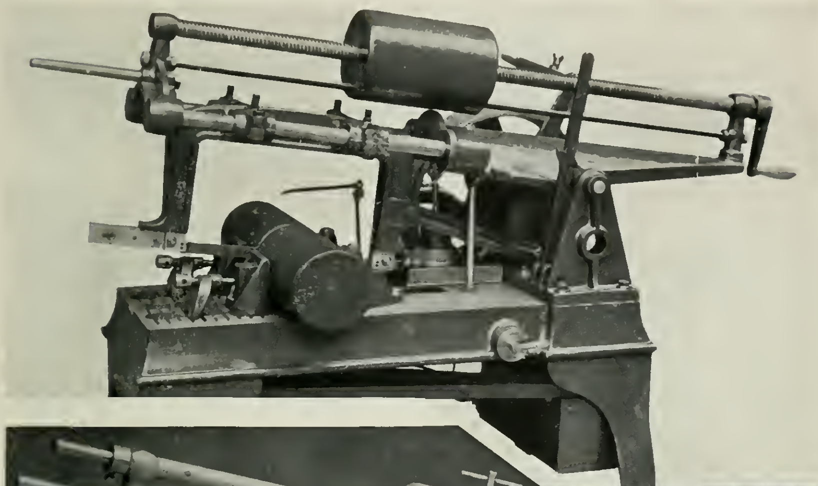


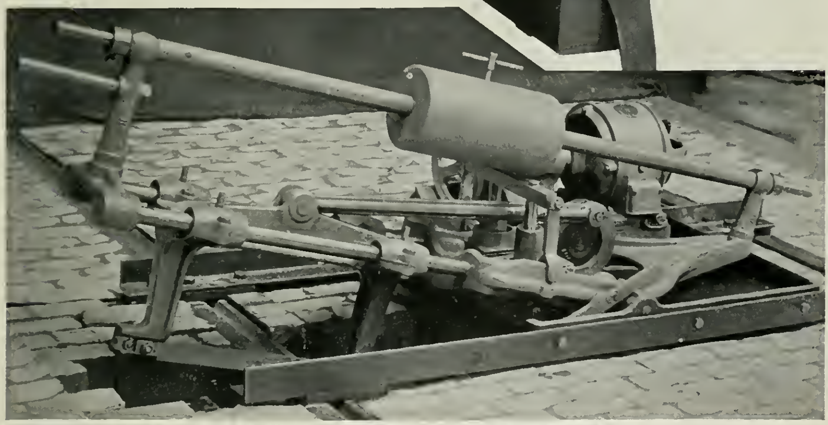
*Lang's lay Rope, New and Worn.*



## STRAIGHT-BLADE SAWING-MACHINES. *Plate 40.*

Figs. 13 and 14. *Single Sawing-Machine, and similar machine for cutting Tramway Rails.*





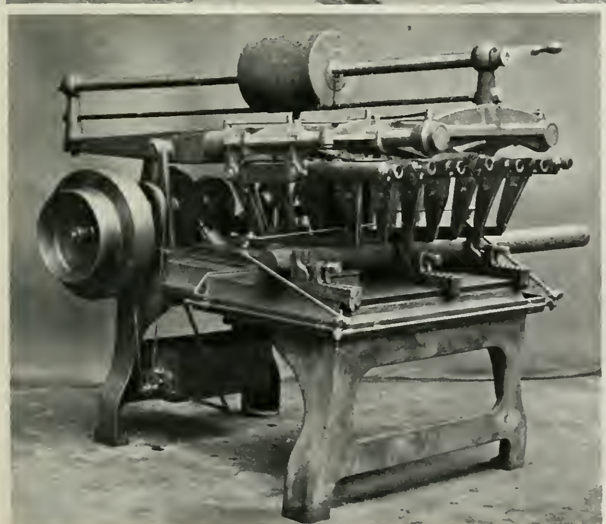


Fig. 15.

*7" Multiple Saw for cutting blanks from $\frac{3}{4}$" thick up to the capacity of the machine.*

*Mechanical Engineers 1912.*

Fig. 19. *Shaping Machine Saw.*

Fig. 17.
*Multiple Saw.*
*Rack Arrangement for cutting blanks from $\frac{1}{16}''$ to $\frac{3}{4}''$ thick.*

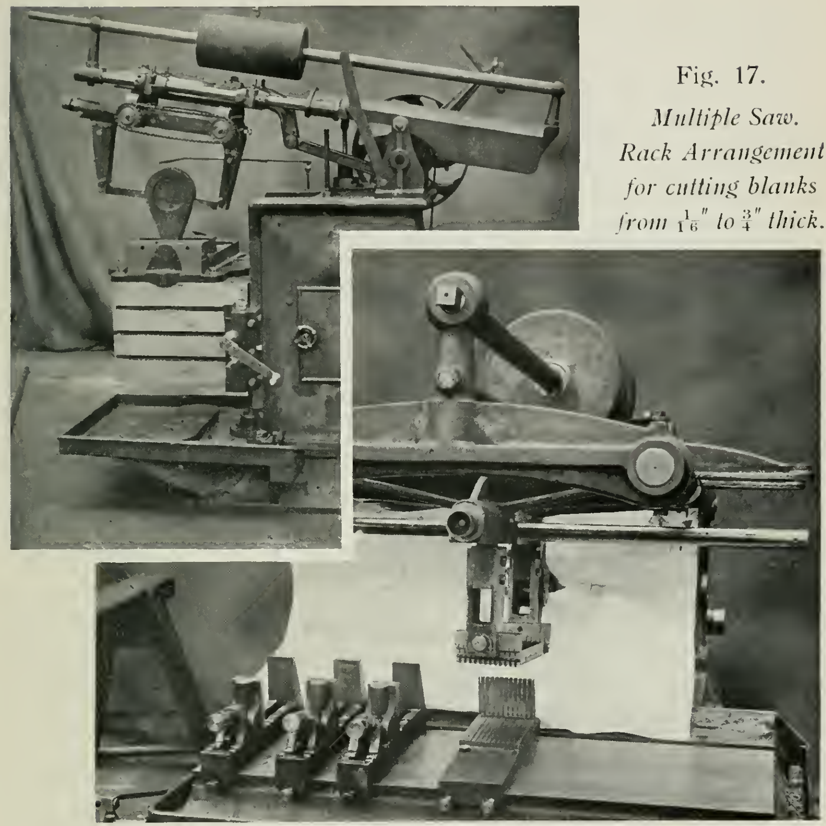
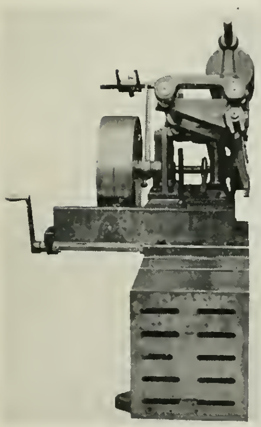
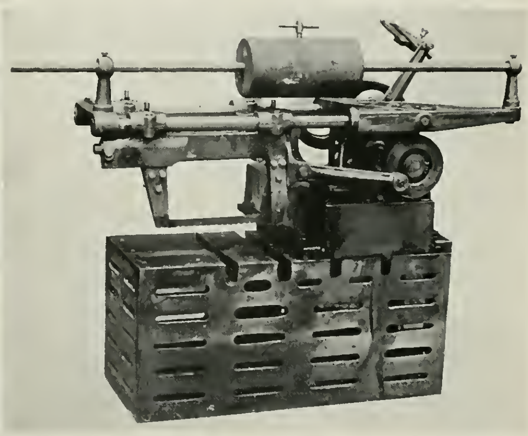
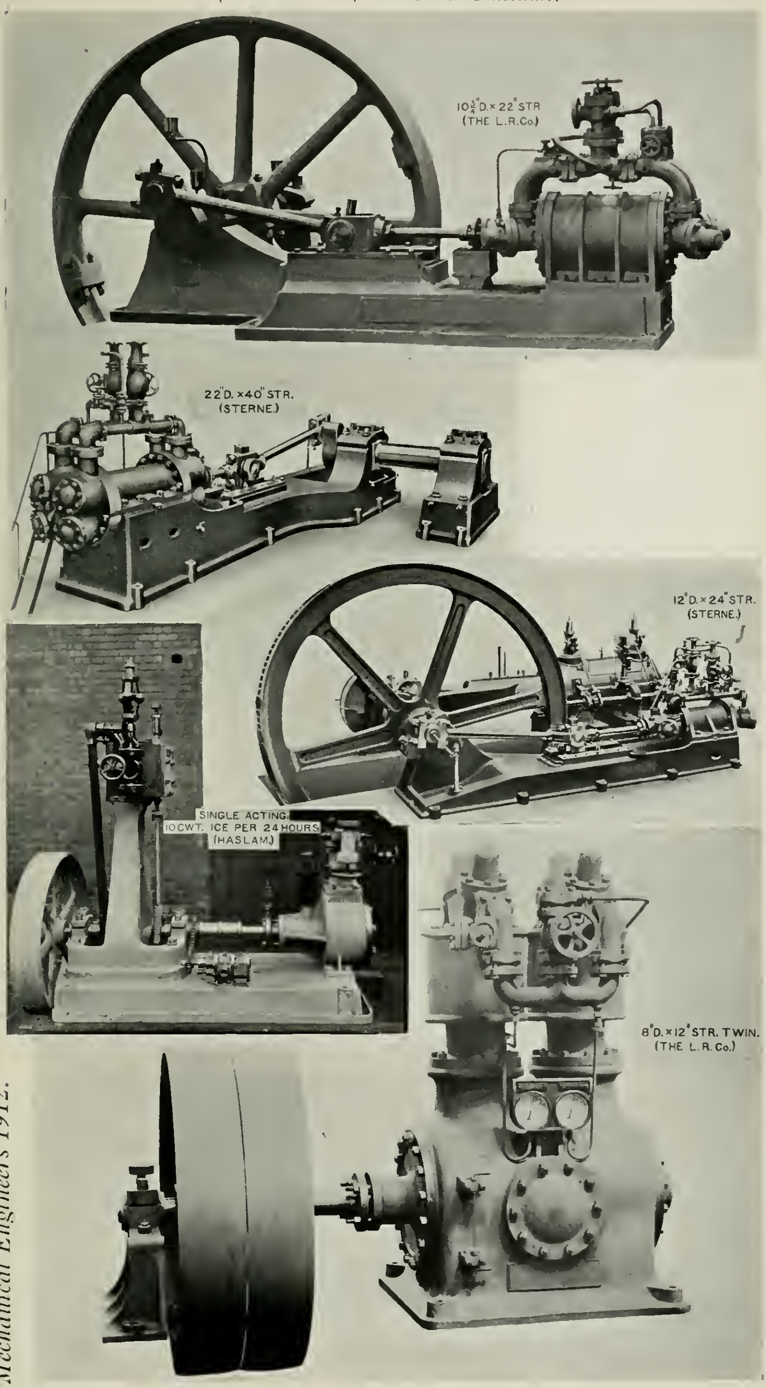


Fig. 21. *Saw for Steel Castings.*



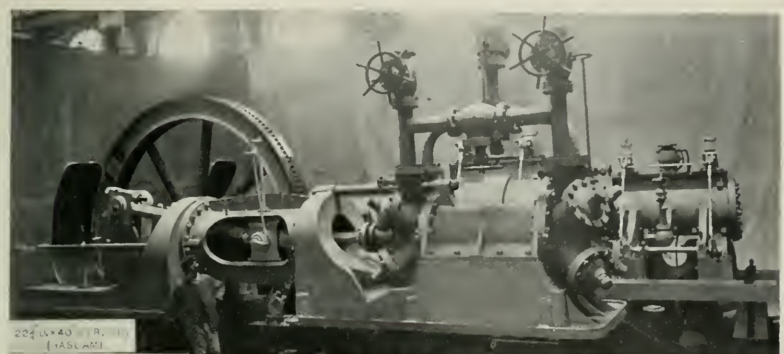


*Horizontal and Vertical Ammonia Compressors, Belt or Steam Driven.*
*(The L. R. Co., Sterne and Haslam.)*

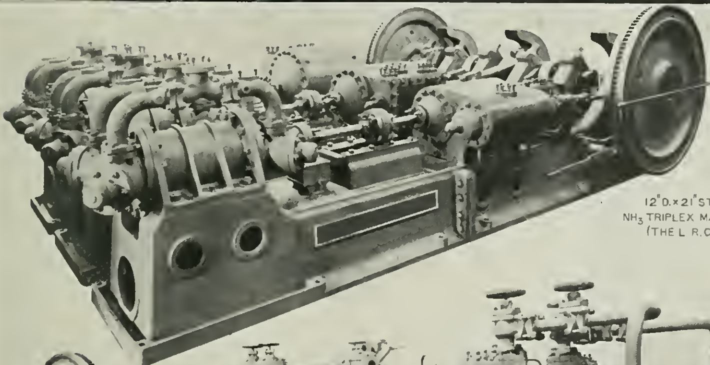



*Mechanical Engineers 1912.*

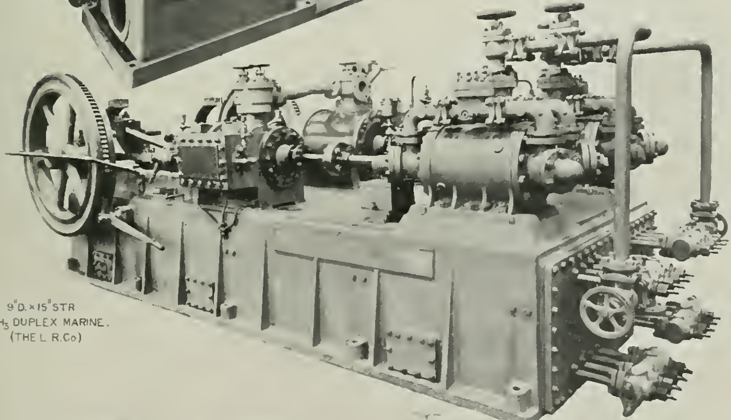
*Ammonia and Carbonic Anhydride Compressors.*
*(Haslam and The L. R. Co.)*



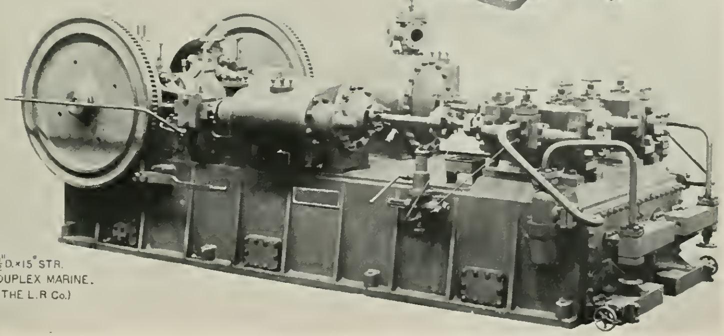
$22\frac{1}{4}$" D. × 40" STR.
(HASLAM)



12" D. × 21" STR.
$NH_3$ TRIPLEX MARINE.
(THE L. R. Co.)



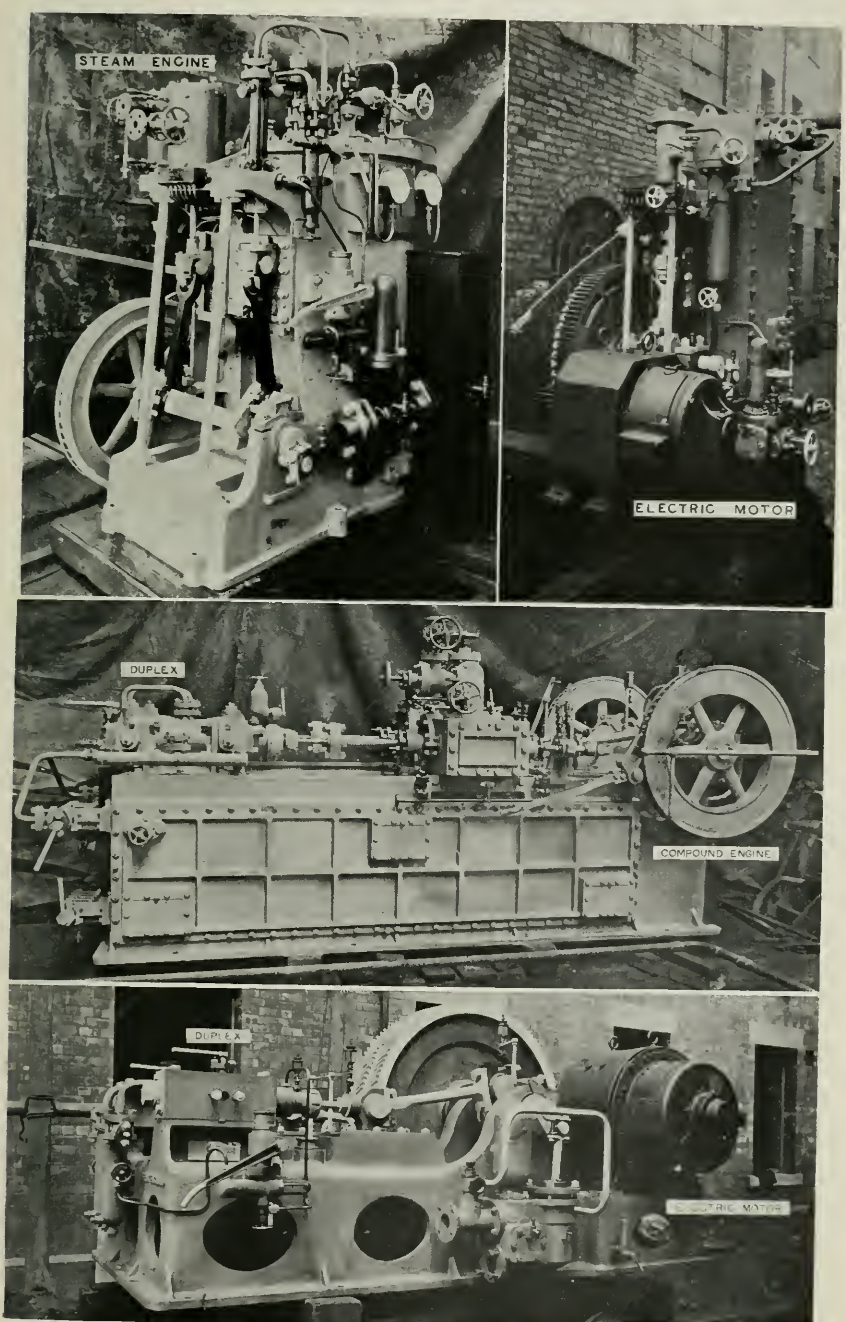
9" D. × 15" STR.
$NH_3$ DUPLEX MARINE.
(THE L. R. Co.)



$4\frac{1}{2}$" D. × 15" STR.
$CO_2$ DUPLEX MARINE.
(THE L. R. Co.)

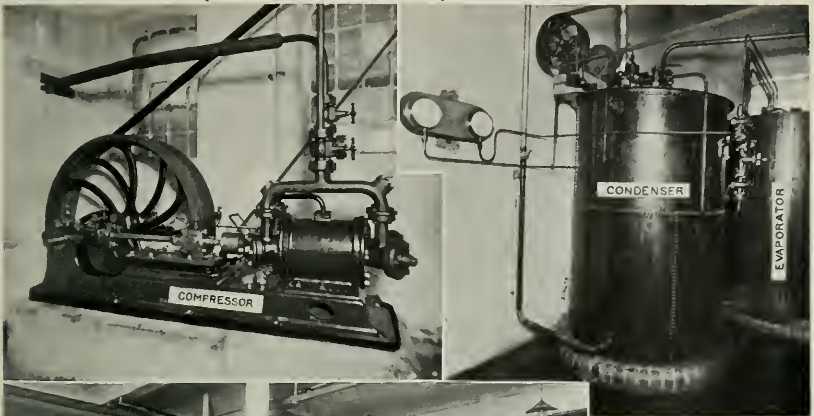
*Mechanical Engineers 1912.*

*Carbonic Anhydride Marine Compressors. (Haslam.)*



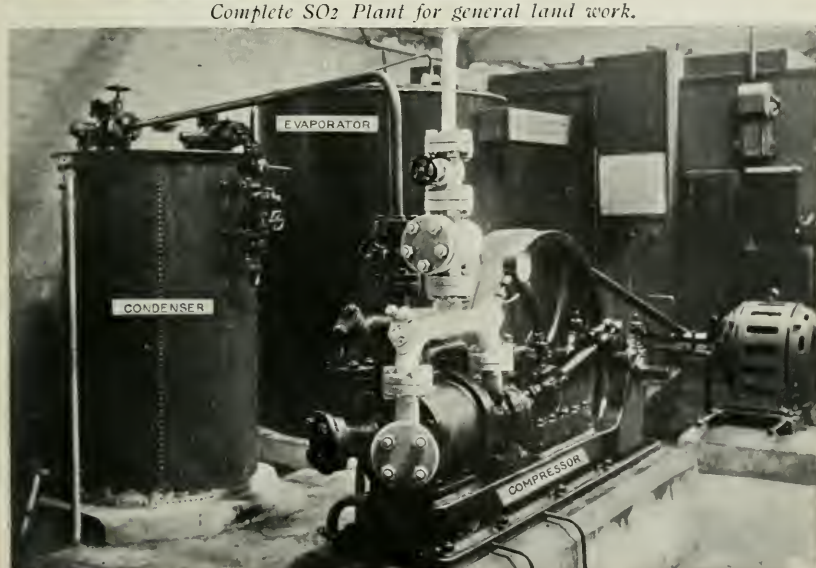

*Mechanical Engineers 1912.*

*Sulphurous Anhydride Compressors. (Douglas.)*




*Dairy, showing Milk cleaner, Pasteurizer, Separator, etc., and Compound Milk-Cooler* (See Fig. 19.) *working in conjunction with above plant.*

*Complete $SO_2$ Plant for general land work.*



*Pl. 46.*

*(Mr. Arthur G. Enock's remarks.)*

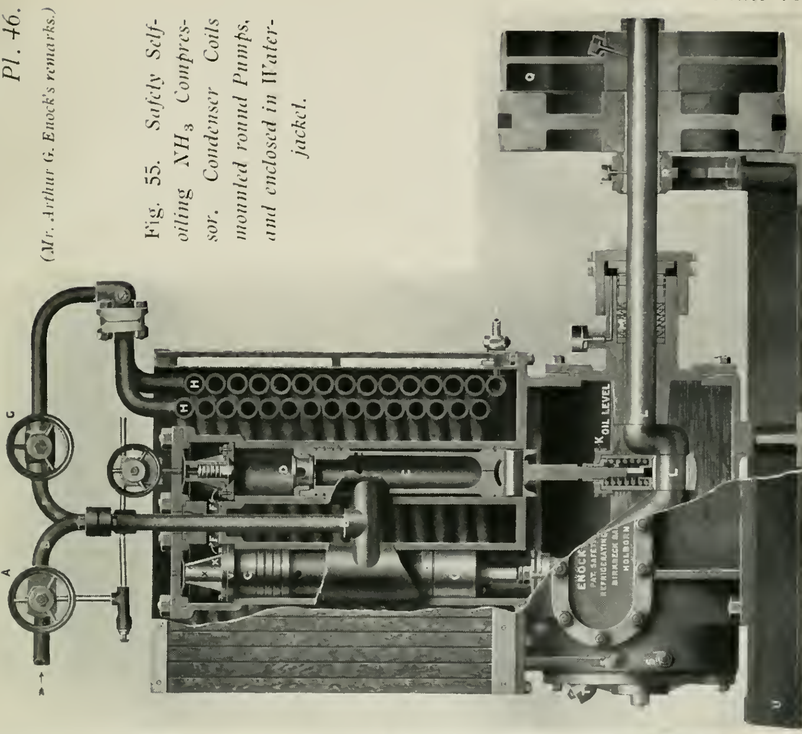
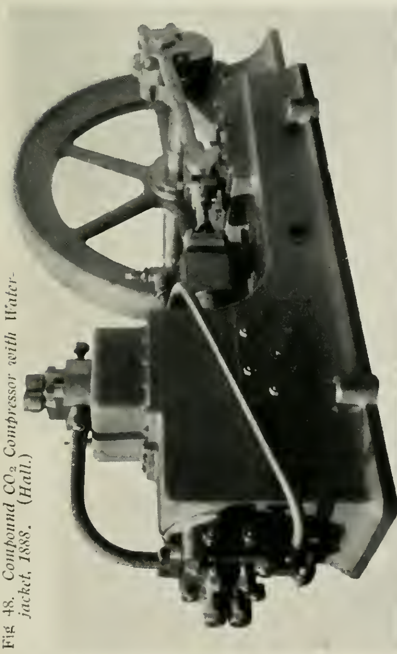



*Fig. 55. Safety Self-oiling $NH_3$ Compressor. Condenser Coils mounted round Pumps, and enclosed in Water-jacket.*



*(Mr. G. C. Hodsdon's remarks.)*

Fig 48. *Compound $CO_2$ Compressor with Water-jacket, 1888. (Hall.)*

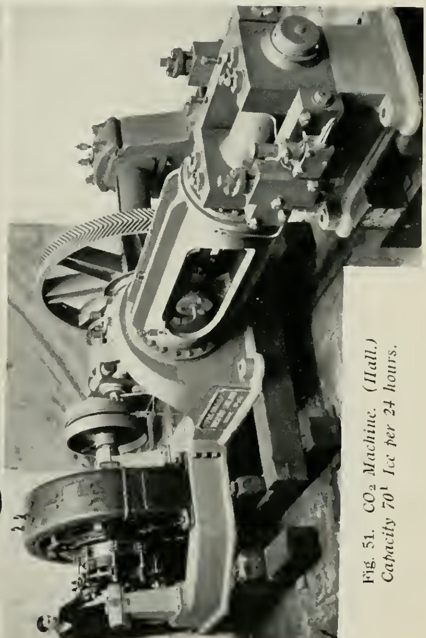


Fig. 51. *$CO_2$ Machine. (Hall.) Capacity 70[t] Ice per 24 hours.*